The New Isuzu 1.8 Liter 4-Cylinder Diesel Engine for the United States Market

Author(s): M. Higashisono, K. Takeuchi and H. Hara

Source: *SAE Transactions*, 1982, Vol. 91, Section 1: 820003—820283 (1982), pp. 384-419

Published by: SAE International

Stable URL: https://www.jstor.org/stable/44631952

820116

The New Isuzu 1.8 Liter 4-Cylinder Diesel Engine for the United States Market

M. Higashisono, K. Takeuchi, and H. Hara
Light Duty Engine Design Dept.
Isuzu Motors Limited
(Fujisawa, Kanagawa, Japan)

FROM ITS FOUNDATION IN 1916, the Isuzu Motors Limited has continuously been making its efforts to offer the vehicles to meet the ever-changing social needs. Current product lines cover all types of vehicles ranging from passenger cars to heavy-duty trucks and busses. The company has been producing all types of automobile engines, both gasoline and diesel too. The company started its production of diesel engines for trucks in the 1930's. Since then the product line has been far expanded, currently, ranging from 0.86-liter 2-cylinder up to 16.8-liter V-12. In 1980, a total of 402,000 units of diesel engines were produced including those for passenger cars.

The Diesel-powered passenger car was first marketed in 1961, which was followed by the models introduced in 1964, 1977 and 1979. In order to answer an ever-intensifying fuel economy need, the company decided to build an all-new small fuel-efficient and high-performance diesel-powered car, the Isuzu Gemini Diesel. This model was first marketed in Japan with favorable acceptance. Its reputation has continued to date. The need for diesel-powered passenger cars has been increasing in the United States too. To meet such market needs, the company decided to come to the U. S. market with the ISUZU I-Mark Diesel through the sales channel of the American Isuzu Motors Inc. an ISUZU affiliate, newly established in 1981. The Isuzu 4FB1 1.8-liter diesel engine is mounted on that vehicle. This engine has also been mounted on a sub-compact car, "Chevrolet Chevette Diesel", of the General Motors. The company has had a business tie-up with General Motors since 1971. The engines to be shipped to the U. S. have been engineered to satisfy

ABSTRACT

The new Isuzu 1.8-liter diesel engine was developed with our special efforts given to quiet operation, light-weight, high performance (SAE 38KW/5000RPM), easy handling (3.5 second pre-heat), meeting the U. S. statutory standards and market requirements, and to excellent fuel economy.

This engine is mounted on the Isuzu I-Mark Diesel and Chevrolet Chevette Diesel which have been on the U. S. market since Spring, 1981. They deliver a fine EPA city fuel economy rating -30% better than their gasoline-powered competitives.

0096-736X/83/9101-0384$02.50

the U. S. requirements, both statutory and operational in field use.

SOCIAL ENVIRONMENTS SURROUNDING DIESEL-POWERED PASSENGER CARS

IN JAPAN - The first production diesel-powered passenger cars were built in the first half of the 1960's. Their good fuel economy was highly welcomed by taxi companies. Those built were, therefore, primarily for taxi service. Diesel-powered taxicabs have been increasing in number. The situation has changed, however. Liquefied Petroleum Gas (LPG) had come up as a more favorable fuel over diesel fuel, tax-wise and as a result price-wise. Privileged position was shifted from diesel to LPG. LPG-powered taxicabs began increasing in number. To make the matter worse, diesel-powered cars were increasingly blamed for their excessive noise and vibration. As a result, the production of diesel-powered passenger cars showed a sharp drop to a level of only less than one hundred units a month. This situation lasted long.

Things changed once again in favor of the diesel. Fig. 1 and 2 show the changes in the

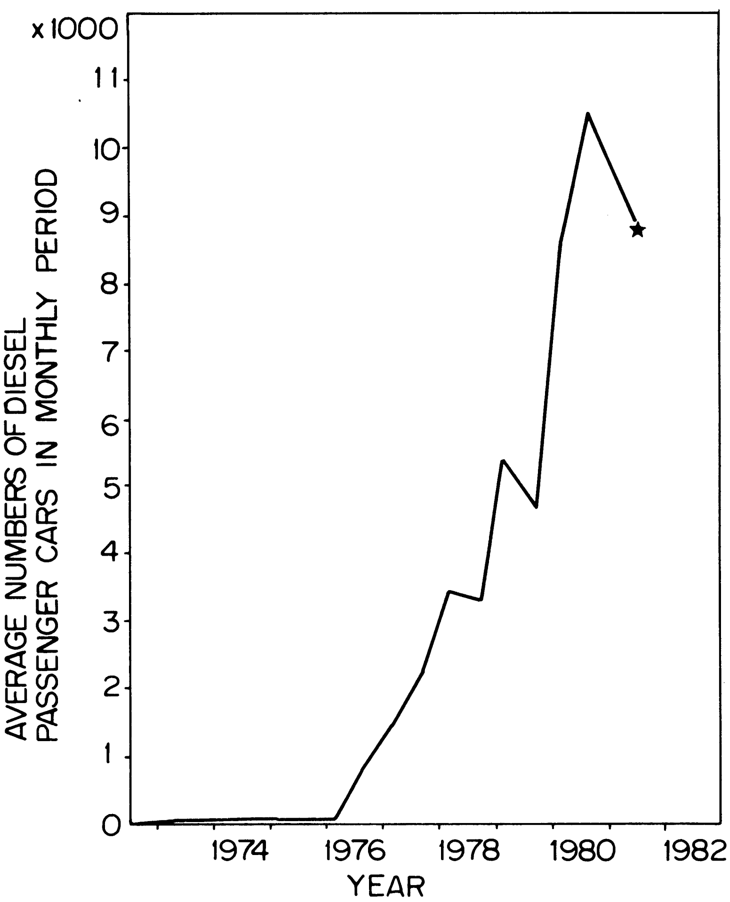


Fig. 1 - Yearly change in number of diesel passenger cars registered in Japan

NOTE: Point with *mark shows the monthly average number of diesel passenger cars registered during period May thru August, 1981

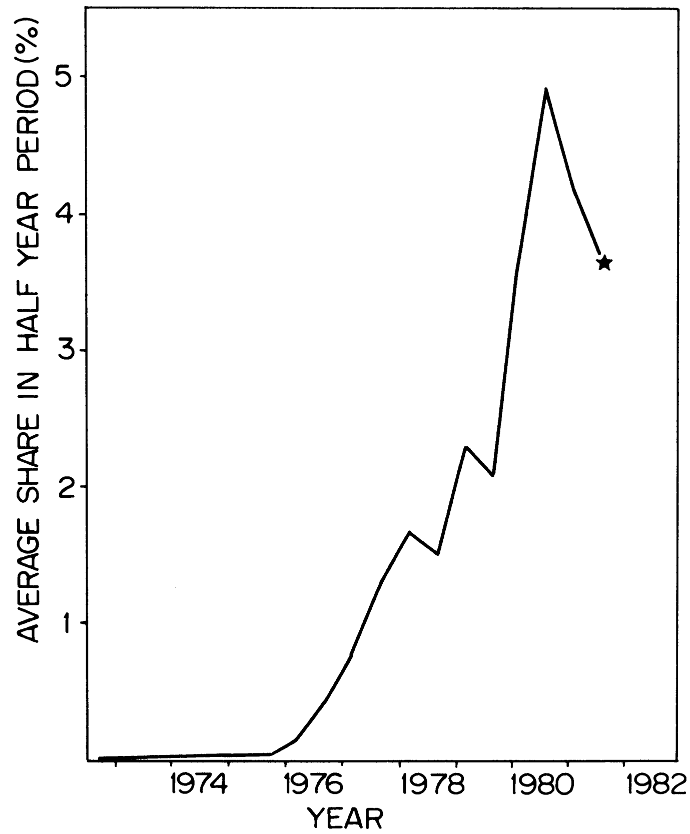


Fig. 2 - Yearly change in share of diesel passenger cars in all passenger cars registered in Japan

NOTE: Point with *mark shows the share of diesel passenger cars registered during period May thru August, 1981

number of diesel-powered passenger cars officially registered and the rate of dieselization in Japan. In 1976, the number of diesel-powered passenger cars rapidly increased. This trend has been further accelerated since that time. Every month in 1980, an average of 10,600 units were built, and the rate of dieselization of the passenger cars came up to a level of 4.9 percent.

We can point out two reasons for such a sudden increase. One is a sharp rise in the fuel costs. Since the fuel crisis in 1974, fuel prices have continued to increase. Current fuel prices are over 3.5 times as high as those experienced in the 1960's, both for gasoline and diesel fuel, as shown in Fig. 3. Speaking of vehicle prices, current prices are approx. 1.7 times as high as those in the 1960's. Thus high fuel prices have forced the Japanese motoring public to be more fuel conscious than ever before, making them diesel-oriented in their passenger car selection. The other is an advanced automotive technology aggressively reflected on the diesel-powered passenger cars. Several drawbacks, once considered rather inherent in the past, such as high level of noise and vibration, insufficient vehicle

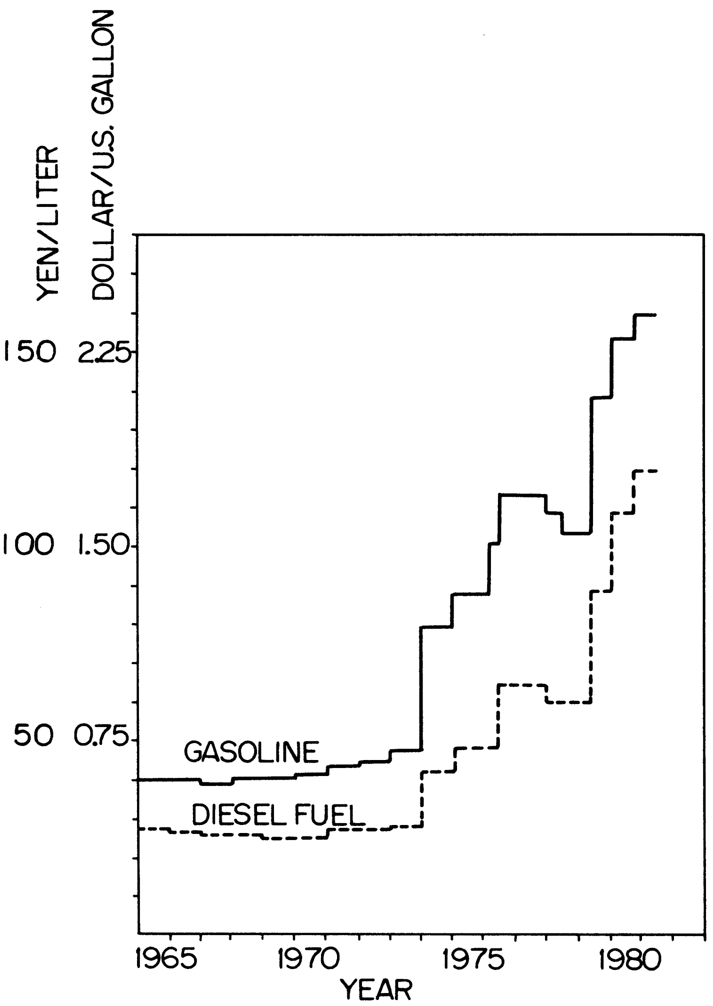


Fig. 3 - Yearly change in automobile-use retail fuel prices in Japan

performance at high speeds, and troublesome operation have been corrected to a considerable extent.

Diesel-powered passenger cars have now come very close to their gasoline-powered counterparts. With these improvements, people are now looking more favorably at them. Table 1 shows the specifications and performance of the Isuzu 1.8-liter diesel engine compared to the Isuzu 2.0-liter diesel engine which was first introduced domestically in 1962. You can find how big the improvements are.

IN THE UNITED STATES - As shown in Fig. 4, the fuel costs in the United States have continued to increase. The same phenomenon has been experienced in Japan. Needs for better fuel economy have been intensified. The Government has also become very sensitive to fuel saving by automobiles. In 1975, the Energy Policy and Conservation Act came into effect. CAFE Standards have been set up. In 1978, the National Energy Policy and Conservation Act came into effect, too.

For attainment of the 1985 CAFE Standard of 27.5 mpg applicable only to gasoline-powered vehicle fleet, the maximum possible extent of vehicle and engine down-sizing and weight reduction as well as the maximum use of innovative automotive technology have to be realized, even if they will be difficult or costly. Dieselization of passenger cars is another method to take up. Automakers and fuel-conscious motoring public now share the same interests with each other. This is why an increasing number of diesel-powered passenger cars have been around as shown in Fig. 5 and 6.

ENGINEERING TARGETS CHALLENGED IN THE DEVELOPMENT OF THE NEW ISUZU 4FB1 1.8-LITER DIESEL ENGINE

The Isuzu Motors was quick to catch the changing social trends which were prevailing both domestically and in the United States, as mentioned earlier, getting its development of a diesel-powered passenger car started which was very close to the gasoline-powered counterpart, performance-wise, operation ease, and improved in fuel efficiency. The development was made based on its long-cultivated and well-proven diesel technology. The U. S. version was engineered so as full to meet the specific local requirements of the United States. For attainment of these targets, our engineering efforts were directed to the realization of the following six factors:

- High-speed, high-performance engine
- Good fuel economy engine
- Light-weight engine
- Low-noise engine
- Easy operation
- To meet the U. S. statutory standards and market requirements

MAJOR SPECIFICATIONS OF THE 4FB1 1.8-LITER DIESEL ENGINE

Diesel-powered passenger cars which had been around in the past fell behind gasoline-powered counterparts in acceleration and the maximum vehicle speeds. This under-performance had long been one of the customers' complaints. In engineering the 4FB1 engine, therefore, its rated maximum speed was designed to 5,000 rpm. For securing high-speed durability and low level fuel consumption, an over-square engine is preferable. Whereas, for good fuel combustion efficiency in diesel engines of smaller displacement, an under-square engine is preferable. After these two sides were fully considered, we adopted a square stroke-bore ratio of 0.98 for this engine. Reduction of the friction loss for less fuel consumption while running at high speeds and reduction of the weight of moving parts for lowered vibration also while running at high speeds were the items of our special emphasis. This target of engine rpm was successfully accomplished. When no load is given, the maximum speed of this engine comes up to as high as 5,500 rpm.

Table 1 - Specifications and performance comparison of new generation 1.8L diesel and old generation 2.0L diesel passenger car

Item		Unit	1.8L Diesel (Domestic)	2.0L Diesel (Domestic)
Date, Market Introduction		——	Oct.'79	Feb.'62
Gross Vehicle Weight		kg	1275	1560
Engine Model		——	4FB1	DL201
Displacement		cm^3	1817	1991
No. of Cylinders		——	4	4
Bore x Stroke		mm	84 x 82	83 x 92
Max. Power (JIS)		KW/rpm	45/5000	40.5/3800
Max. Torque (JIS)		N.m/rpm	110/2000	121/2200
Fuel Economy - 60 km/hr		km/L	29	18.5
Performance	0-100 km/hr(62MPH)	sec	20.3	27.6
	Max. Speed	km/hr	144	105

Maximum engine output is SAE 38KW at 5,000 rpm. Maximum engine torque is SAE 97.6 N-m at 2,000 rpm. Dry engine weight is 174 Kg. Table 2 and 3 show the detail specifications of this engine and the vehicle on which this engine is mounted which is compared to those of gasoline counterparts for reference. The outside view drawings and section of drawings of the 4FB1 engine are shown in Fig. 7,8,9,10, 11 and 12.

DESIGN FEATURES OF MAIN ENGINE COMPONENTS AND RELATED SYSTEMS

CYLINDER HEAD AND COMBUSTION CHAMBER - Cylinder head (Fig. 13) is made of special cast iron, having the cross-flow type independent ports, exhaust port on the left side and intake port on the right side. Drilled holes between valve seats are designed as engine coolant passages. Valve seat inserts are made of special heat resisting steel, both exhaust and intake.

Combustion chamber has a RICARDO Comet V type swirl chamber. The configurations of the swirl chamber and main combustion cavity in the piston head and the injection nozzle position angle are best engineered so that a well-balanced engine performance can be secured in terms of engine output at any engine speed ranging from low to high, exhaust smoke, exhaust emissions, noise and engine startability. See Fig. 11.

CYLINDER HEAD BOLTS AND GASKETS - Cylinder head bolts, 13mm in diameter, are 4 per cylinder, totaling 10. Cylinder head gasket is of the laminated type. So as sufficiently to ensure the sealing effect of bore grommets, the thickness distribution of the stainless steel used, configuration of between-plate, and the material, configuration and thickness of rubber rings used in the holes, both for coolant and for lubricant passages, were all carefully examined and determined to the specifications most appropriate. See Fig. 14.

By the use of the thinner stainless steel which is laminated, the variation in the thickness of the production gasket was minimized, contributing to a decreased dead volume of the combustion chamber and also a decreased variation of the engine compression ratio. As for the cylinder head bolt layout, the 4FB1 diesel shares

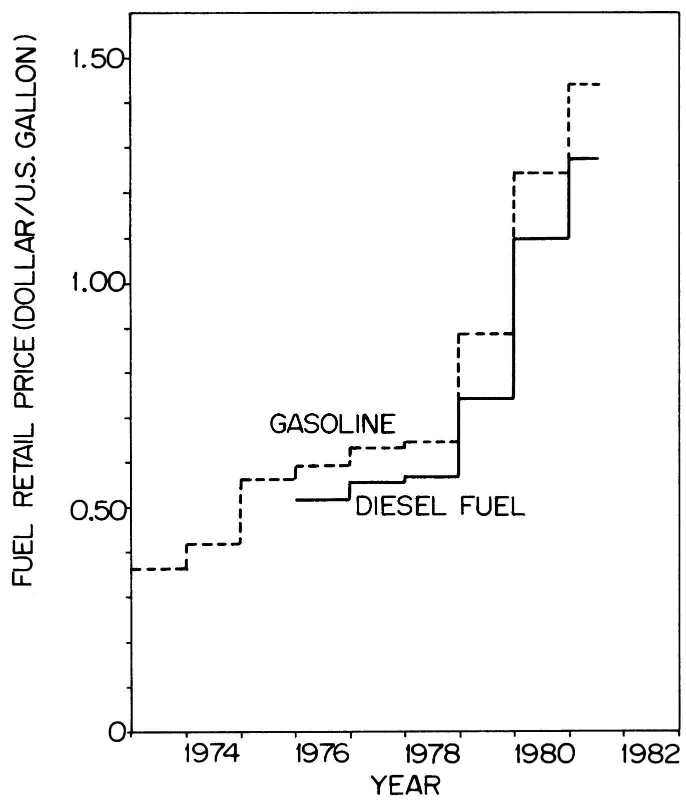


Fig. 4 - Yearly change in automobile-use fuel retail price in U.S.A.

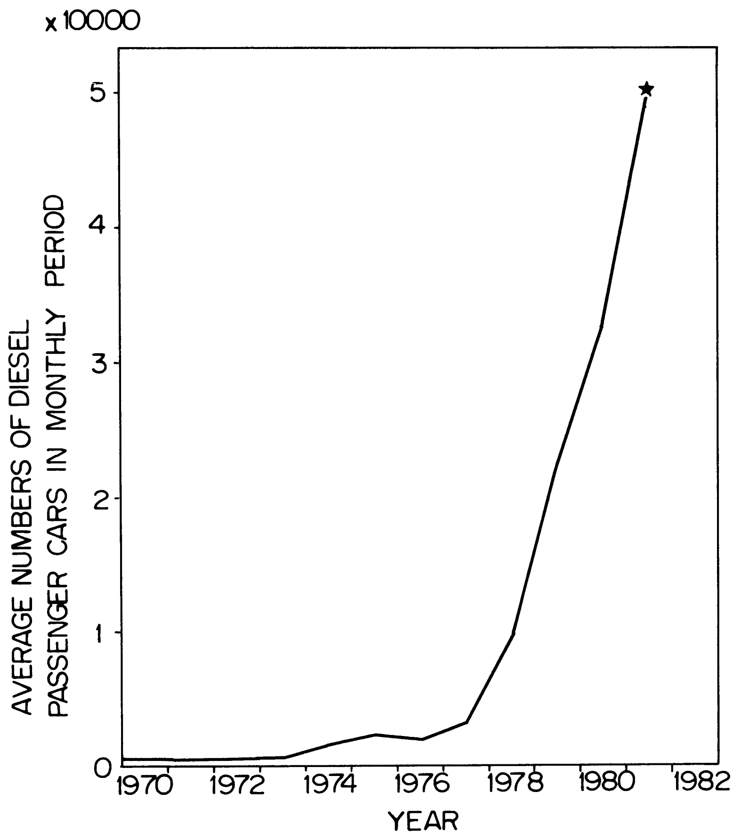


Fig. 5 - Yearly change in number of diesel passenger cars registered in U.S.A.

NOTE: Point with *mark shows the monthly average number of diesel passenger cars registered during period January thru June, 1981

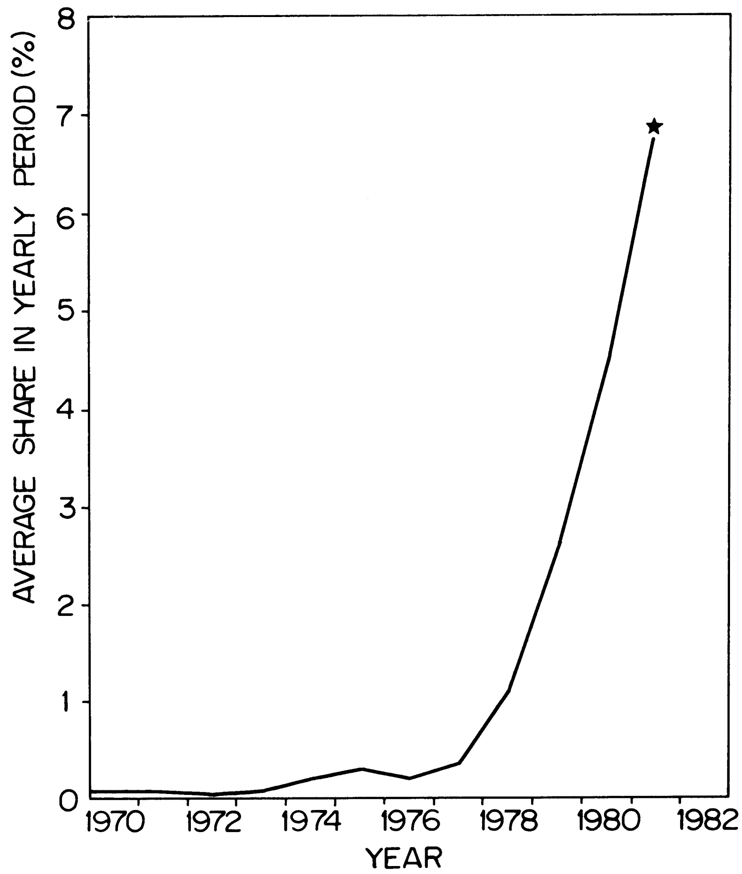


Fig. 6 - Yearly change in share of diesel passenger cars in all passenger cars registered in U.S.A.

NOTE: Point with *mark shows the share of diesel passenger cars registered during period January thru June, 1981

it with its gasoline counterpart, contributing to an increased line productivity.

CYLINDER BLOCK - For weight reduction and down-sizing of the 4FB1 diesel, a best possible commonization was made with its gasoline counterpart in the machining work of cylinder blocks. Included are:

- Cylinder bore
- Cylinder head bolt screw hole
- Crank centerline, with the same main bearings used
- Lower face
- Oil pump rotor housing integrated
- Transmission bolt pattern

Cylinder block (Fig. 15) is made of special cast iron. No cylinder liner is used. Hardness of the surface of the cylinder bore is increased by the adoption of the special cast iron and the special casting method. Adoption of optimum bore surface roughness and cross-hatching angles results in less consumption of engine oil and less volume of blow-by gas as well as a decreased wear of piston rings. Water ducts are engineered in the water jacket portions, ensuring a uniform flow of engine coolant to the cylinders. The rotor housing of the oil pump is integrated in the front wall of cylinder block.

Table 2 - Major specifications of Isuzu 1.8L diesel and 1.8L gasoline engine for the U.S. market

ITEM	UNIT	1.8 L DIESEL	1.8 L GASOLINE
ENGINE MODEL	--	4FB1	G180Z
CYCLE	--	4	4
NO. OF CYLINDER	--	4	4
CYLINDER ARRANGEMENT	--	IN-LINE	IN-LINE
BORE x STROKE	mm	84 x 82	84 x 82
DISPLACEMENT	cm^3	1817	1817
COMBUSTION CHAMBER	--	SWIRL CHAMBER	HEMISPHERICAL
COMPRESSION RATIO	--	22.0	8.5
MAX. OUTPUT (SAE)	kW/rpm	38/5000	58/4800
MAX. TORQUE (SAE)	N.m/rpm	97.6/2000	129/3000
WEIGHT	Kg	174	141
LENGTH x WIDTH x HEIGHT	mm	699 x 574 x 675	695 x 609 x 581
COOLING SYSTEM	--	WATER COOLED	WATER COOLED
CYLINDER HEAD DESIGN	--	O.H.C., CROSS-FLOW	O.H.C.,CROSS-FLOW
FUEL SYSTEM	--	BOSCH DISTRIBUTOR PUMP, VE TYPE	CARBURETOR

Table 3 - Isuzu 1.8L diesel light duty vehicle specifications for 1982 model year for U.S. market

VEHICLE NAME	ISUZU I-MARK DIESEL								
VEHICLE MODEL	PD-1			PD-2		PD-3		PD-4	
BODY STYLE	COUPE			SPORT COUPE		SEDAN		SPORT SEDAN	
* TRANSMISSION	M-1C (M-4)	M-2E (M-5)	A-3 (AT)	M-2E	A-3	M-2E	A-3	M-2E	A-3
INERTIA WEIGHT (LBS)	2,500						2,750	2,500	2,750
EQUIVALENT TEST WEIGHT (LBS)	2,500	2,625					2,750	2,625	2,750
FRONTAL AREA (FT^2)	18.4					18.7			
FRONTAL AREA HP	9.2			9.6		9.4		9.8	
TEST HORSE-POWER	8.8	9.2		9.6		9.4		9.8	
TIRE SIZE	155SR -13	155SR-13 175/70SR-13		175/70SR-13		155SR-13 175/70SR-13		175/70SR-13	
AXLE RATIO	3.154	3.308	3.583	3.308	3.583	3.308	3.583	3.308	3.583
N/V RATIO	48.2	43.3 43.1	54.8 54.6	43.1	54.6	43.3 43.1	54.8 54.6	43.1	54.6
A/C INSTALLED	LESS THAN 33%								

NOTE: * TRANSMISSION CONFIGURATION

TRANSMISSION CODE	1st	2nd	3rd	4th	5th
M-1C	3.47	1.99	1.36	1.00	-
M-2E	3.79	2.18	1.42	1.00	0.86
A-3	2.45	1.45	1.00	-	-

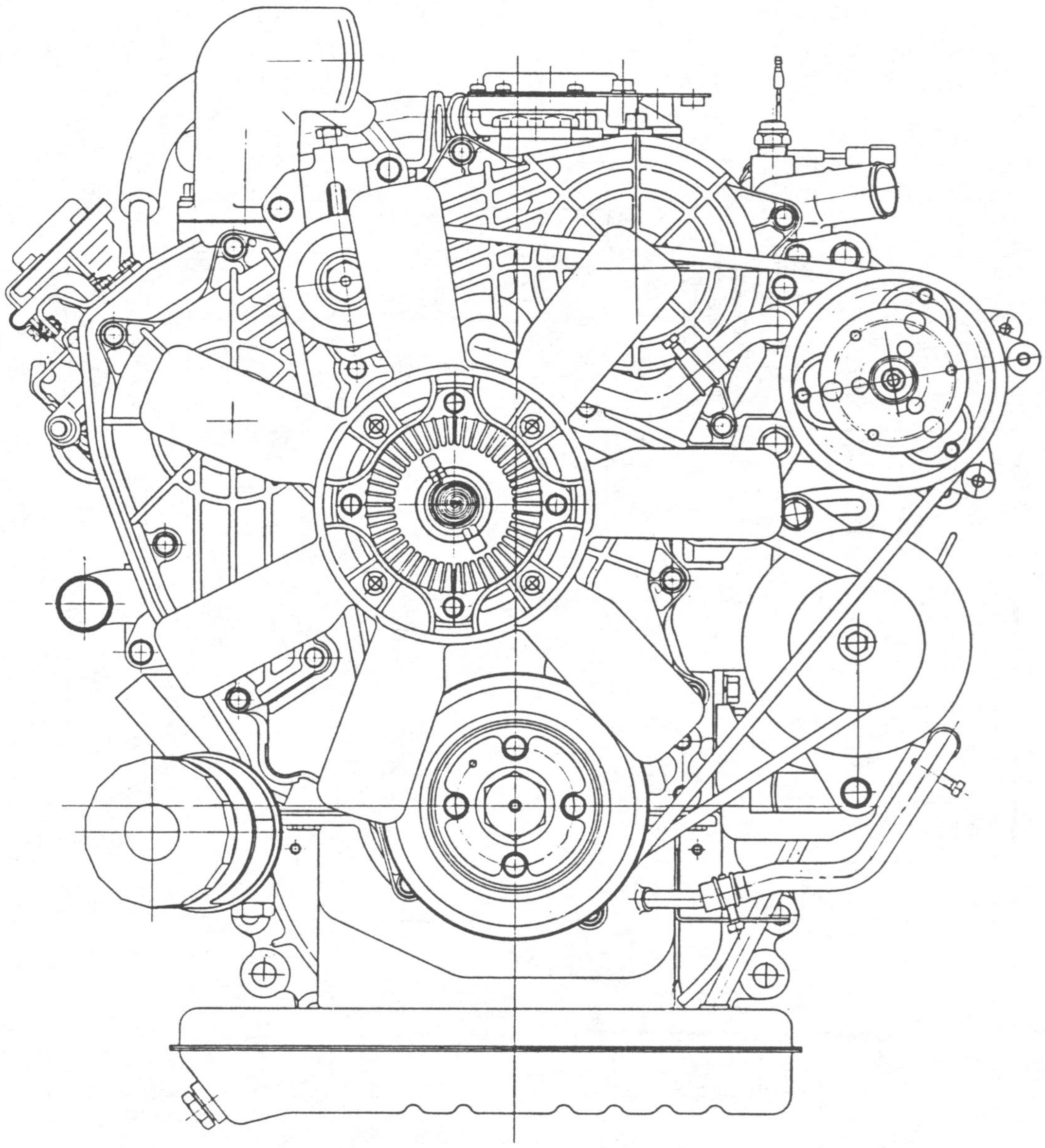


Fig. 7 - Front view of 4FB1 engine

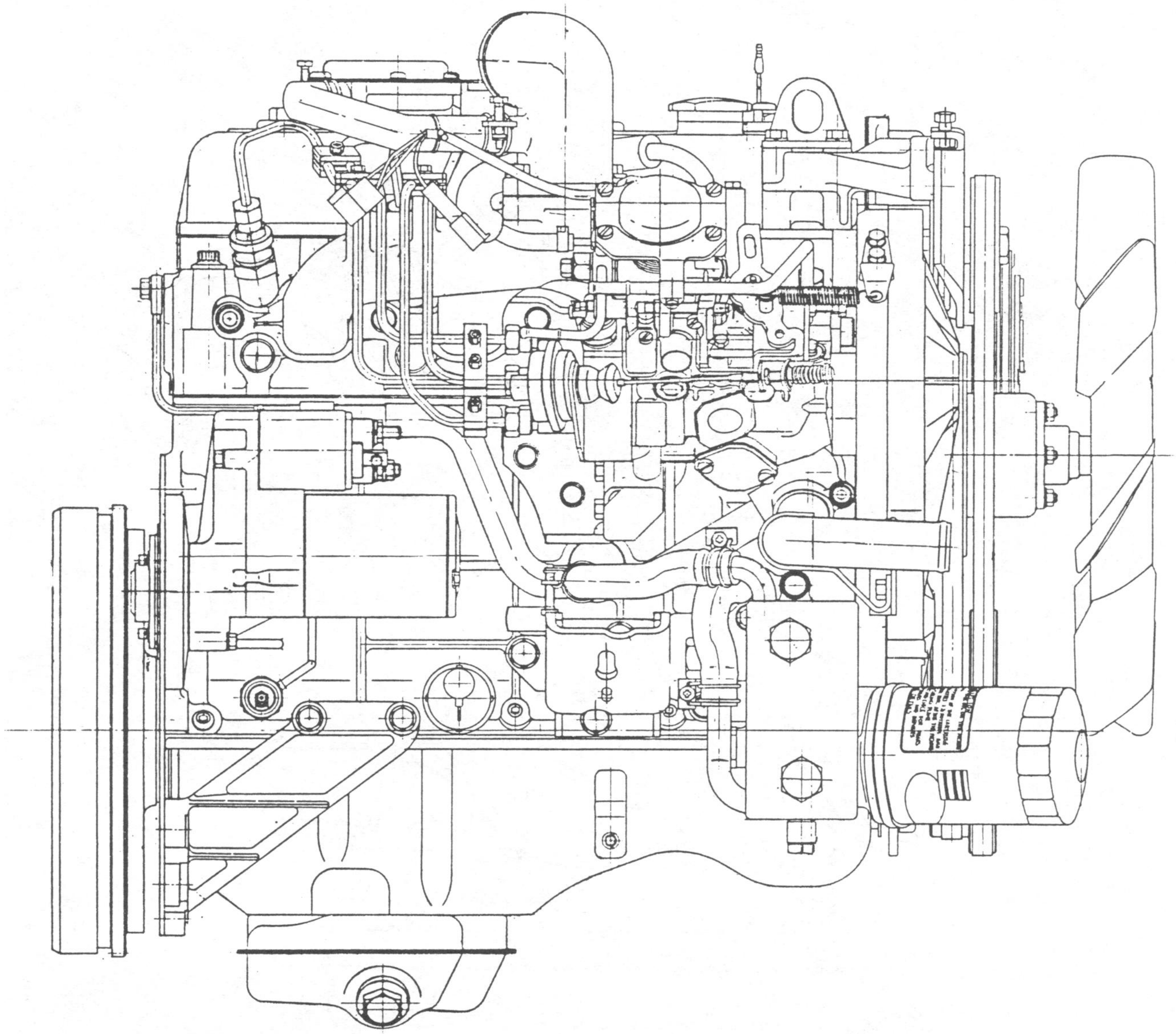


Fig. 8 - Right side view of 4FB1 engine

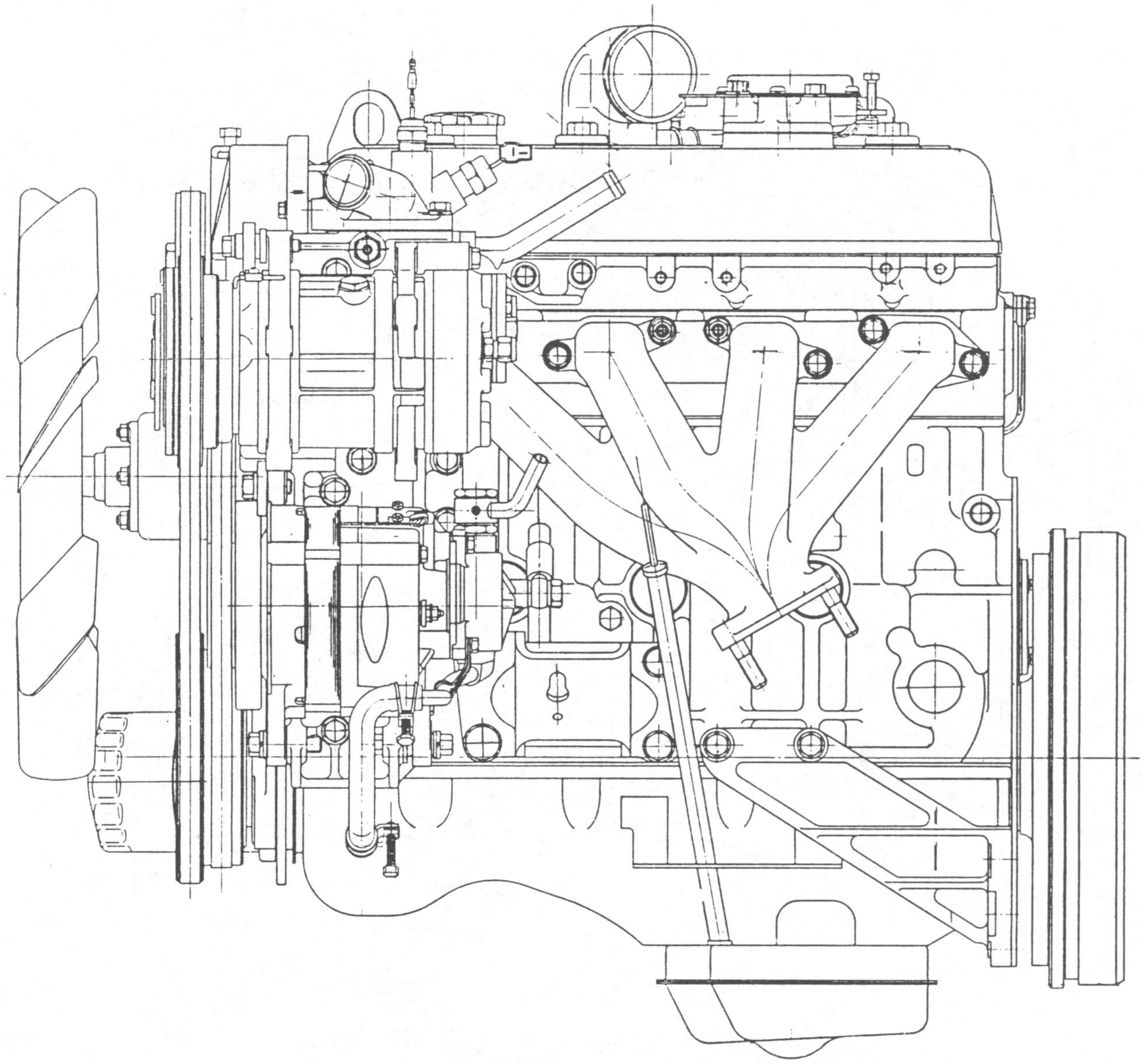


Fig. 9 - Left side view of 4FB1 engine

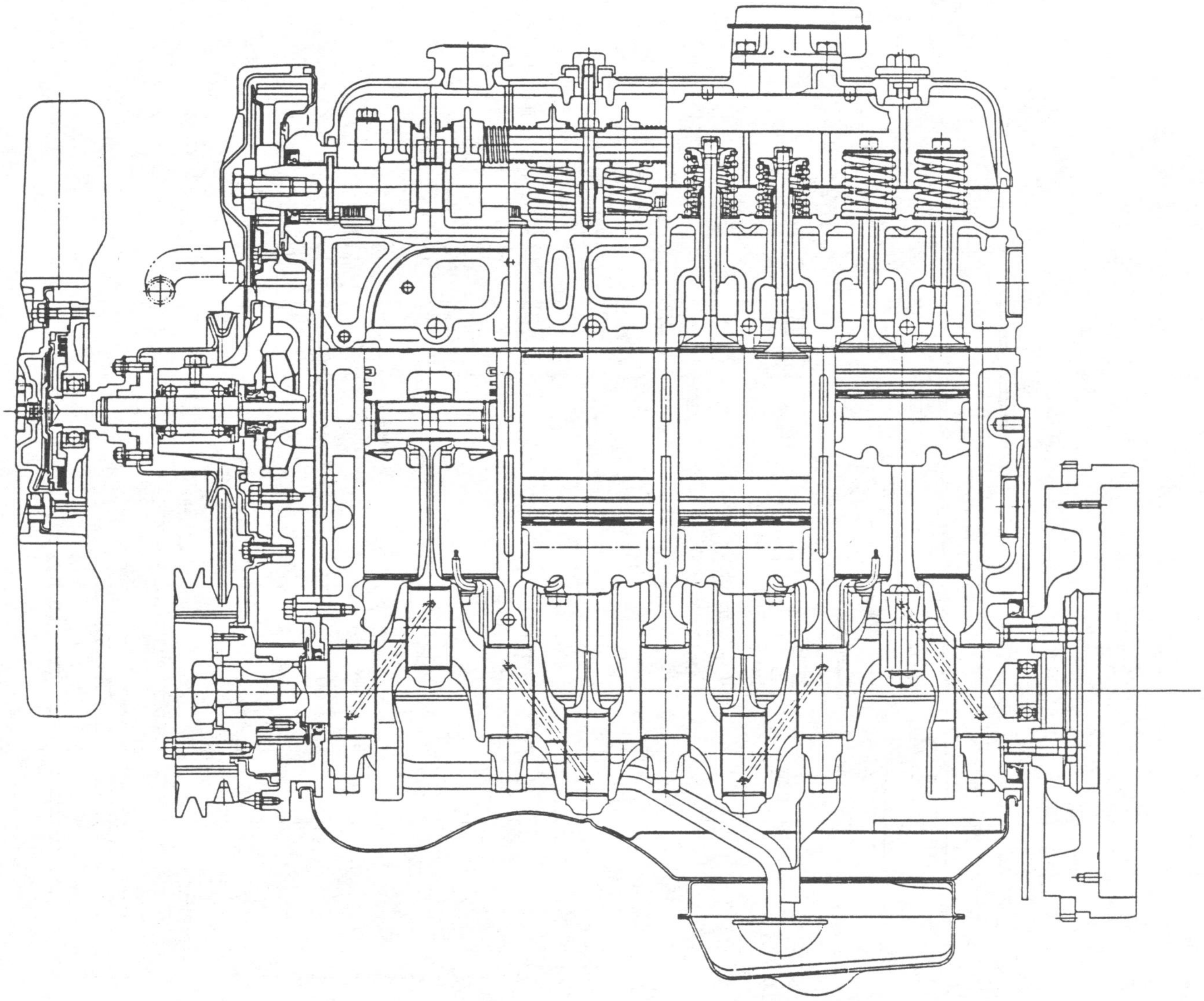


Fig. 10 - Longitudinal section of 4FB1 engine

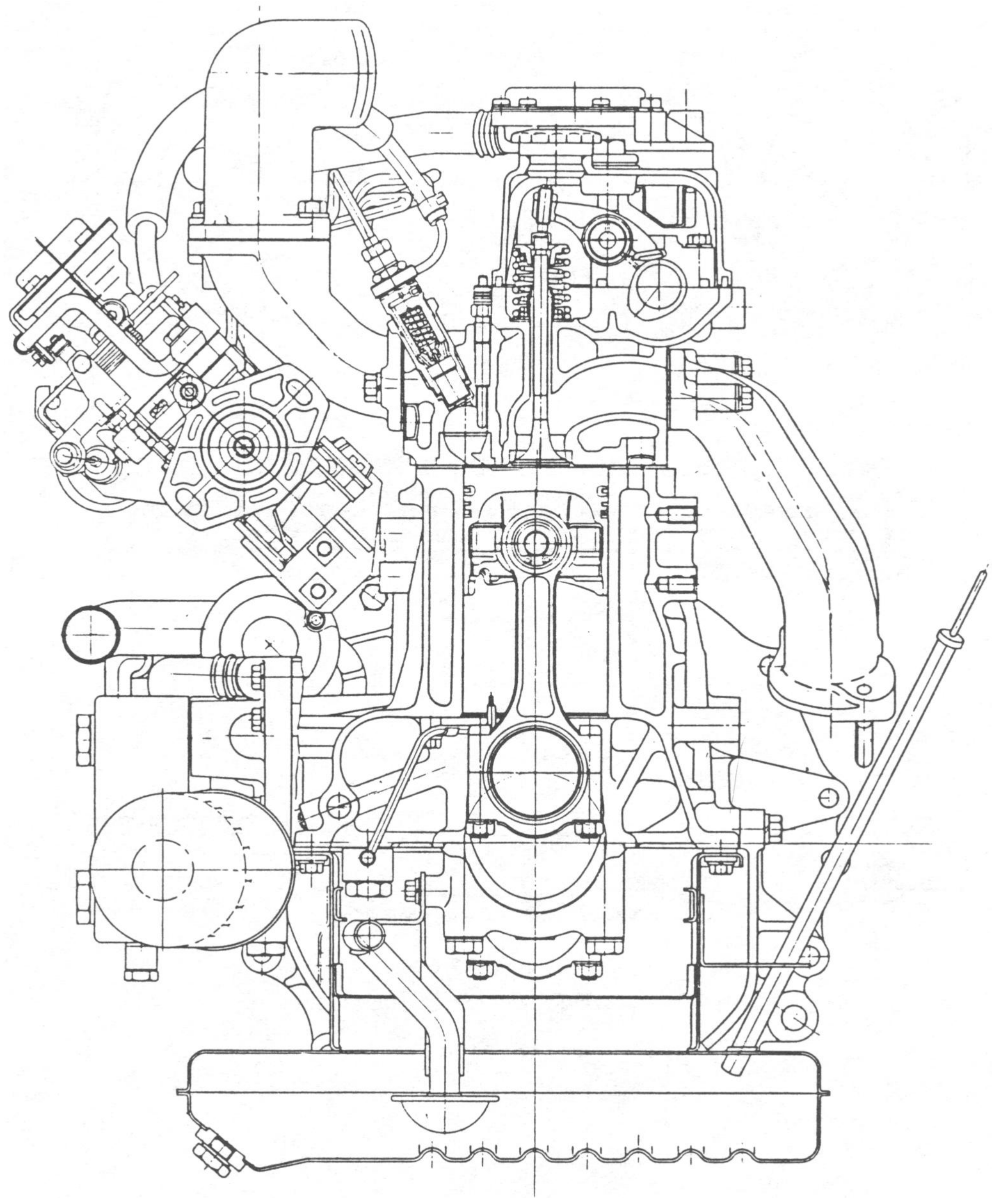


Fig. 11 - Cross section of 4FB1 engine

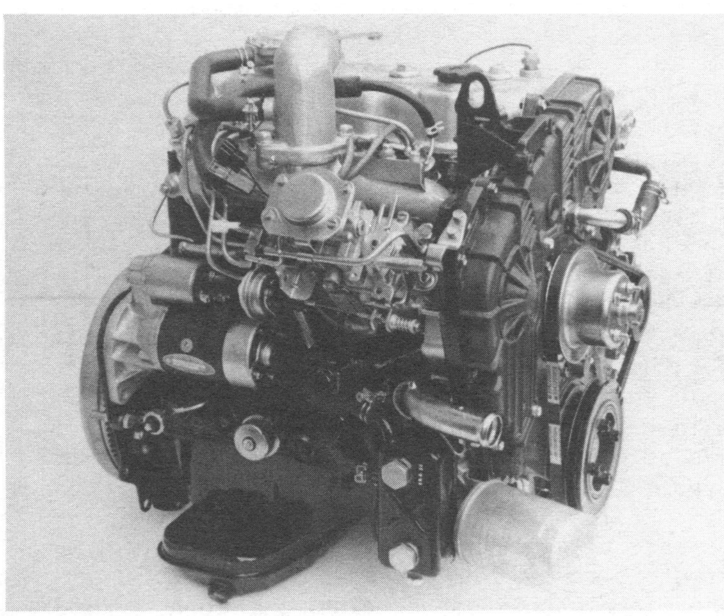


Fig. 12 - Outside view of 4FB1 engine

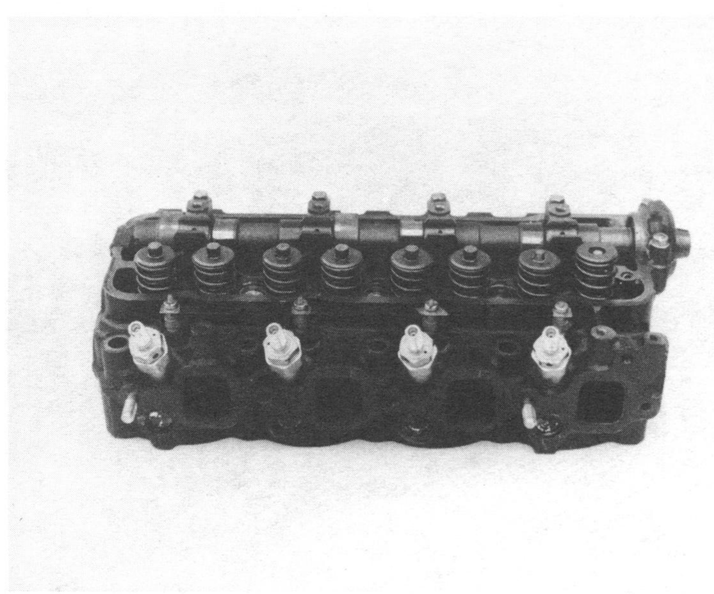


Fig. 13 - cylinder head

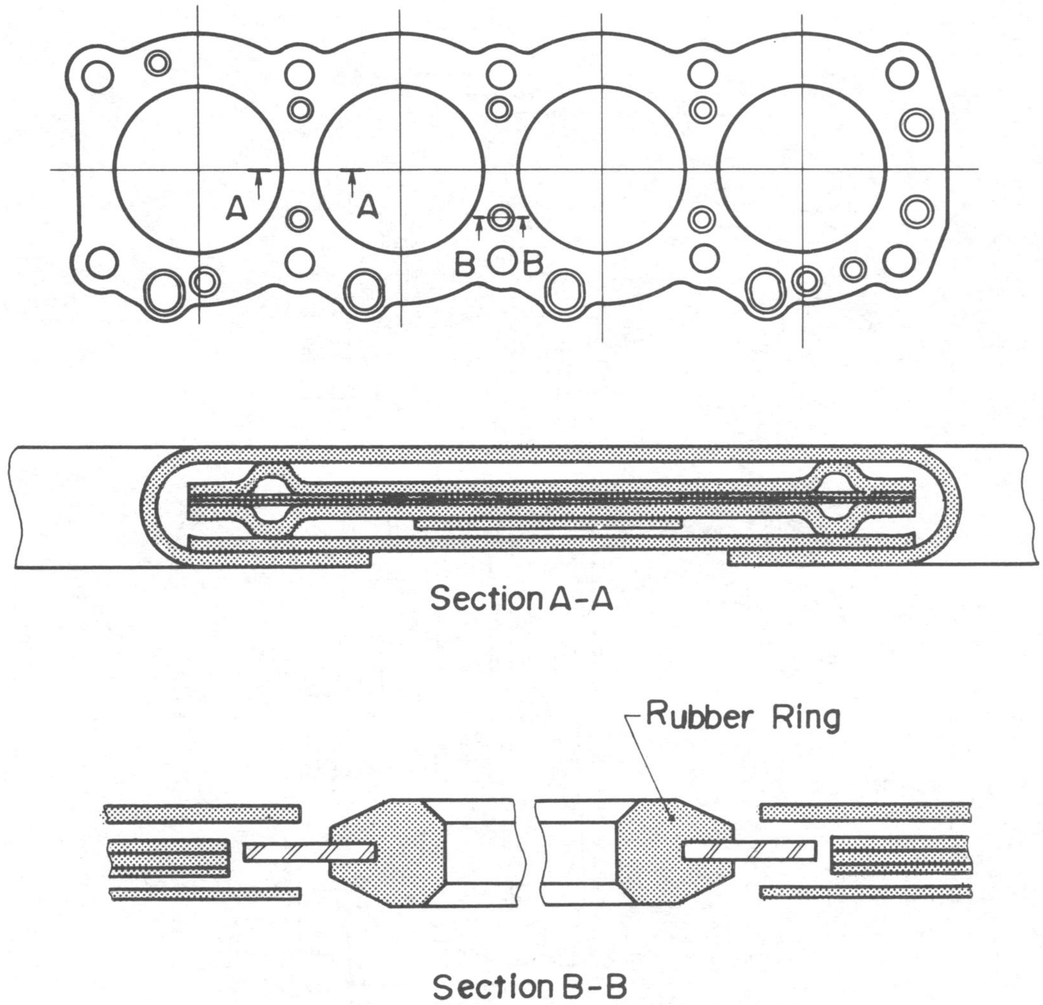


Fig. 14 - Cylinder head gasket

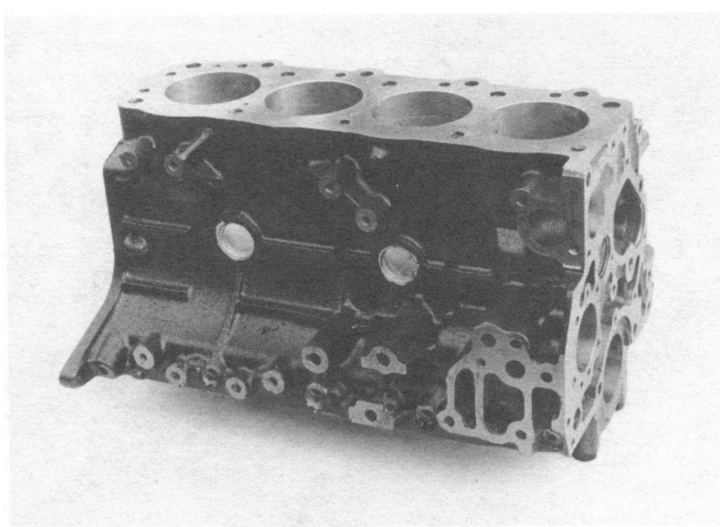


Fig. 15 - Cylinder block

CRANKSHAFT AND CONNECTING ROD - Crankshaft (Fig. 16) is of 5-bearing, 4-balancer type and made of carbon steel. So as to increase the strength against fatigue and wear, a softnitriding surface treatment is applied on the crankshaft. Connecting rod (Fig. 17) is made of special alloy forging. Reamer bolts are used for connecting the rod to the rod cap. Most appropriate material and hardness were selected for these bolts so that sufficient tightening with the nuts can be ensured at all times. Both main bearing and connecting rod big-end bearing are of plain 3-layer copper-lead-indium flashed type.

PISTON AND PISTON RINGS - Piston (Fig. 17) is made of aluminum alloy casting. A steel strut is casted in the piston. This is an autothermic piston whose external sectional profile is designed so that the piston clearance is

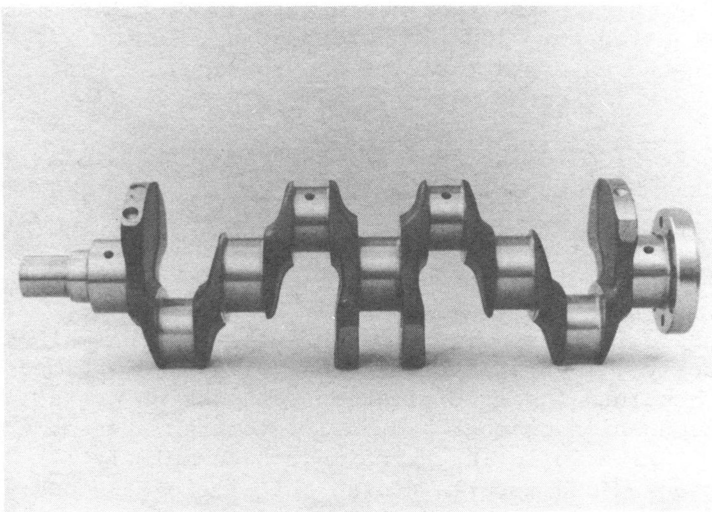


Fig. 16 - Crankshaft

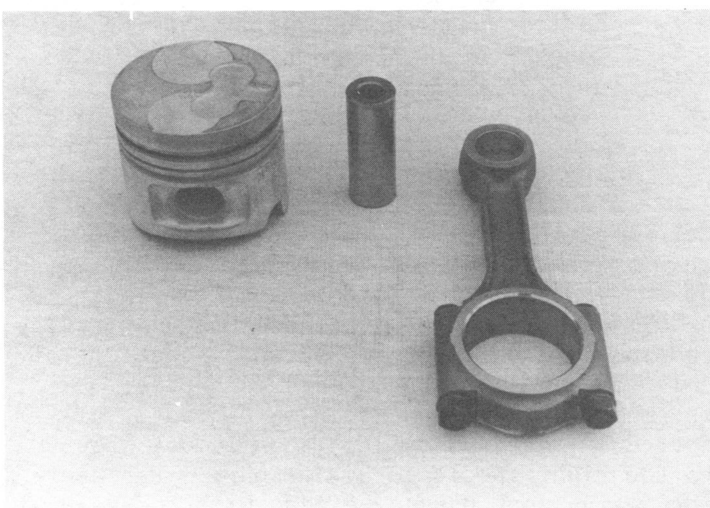


Fig. 17 - Connecting rod and piston

kept to the minimum, which, as a result, contributes greatly to a reduction of the engine noise attributable to the piston slapping. Under the bed of the top ring groove, a Ni-Resist wear-resistant ring is cast in, contributing to an increase in the wearability of the ring groove.

Piston rings consist of two compression rings and one oil-control ring. The first compression ring is barrel-faced and hard chrome-plated. The second compression ring is taper-faced and also hard chrome-plated. Oil-control ring incorporates a coil expander. Both surfaces of the oil control ring, outside and inside, are hard chrome-plated.

VALVE TRAIN AND TIMING TRAIN - Camshaft is made of special alloy cast iron. On its surfaces other than the journals, phosphate treatment is applied. On the cam lobes, chilled treatment is applied. Increased hardness and wearability are secured by this treatment. Rocker arm is made of special alloy forging. On the cam contact surfaces, sintered pad is brazed, ensuring the sufficient hardness and surface roughness of the contact surfaces. Excellent wearability (against scuffing or pitting) is obtained.

Valves, both exhaust and intake, are made of heat-resistant steel. On the valve stems, hard chrome plating is applied, and valve seats are reinforced by Stellite, contributing to the prevention of valve sticking and wear. On the valve ends, sintered caps are mounted, improving their wearability. As for the timing belt, its teeth have the base width of 30mm, and the number of its teeth is 134. The timing belt adopted is made of glass fiber cord and has the tooth profile corresponding to UNIROYAL type 'B'. Timing layout is shown in Fig. 18.

INTAKE AND EXHAUST SYSTEM - Intake manifold is made of aluminum alloy, whereas, the exhaust manifold is made of standard cast iron. The air cleaner incorporates a square-type viscous element therein, ensuring sufficient dust catching capability. Its volume is as big as 7 liters, contributing greatly to a decrease in the air intake noise.

FUEL SYSTEM - Fuel injection pump is Bosch VE type manufactured by the Diesel KiKi Company. Plunger diameter is 8 mm. Governor is of half all speed type, whose characteristics are determined so that it can ensure acceleration feel equivalent to that of gasoline-powered vehicles. Fuel injection pump of VE distribution type features several advantages. They are: compact, small-sized and light-weight, automatically-increased fuel supply at engine start, high-speed engine running, no reverse running in any instance, fuel supply cut-off by turning off engine key, no maintenance service required by virtue of the lubrication by fuel flow through, etc. In addition, mounting of a tachometer

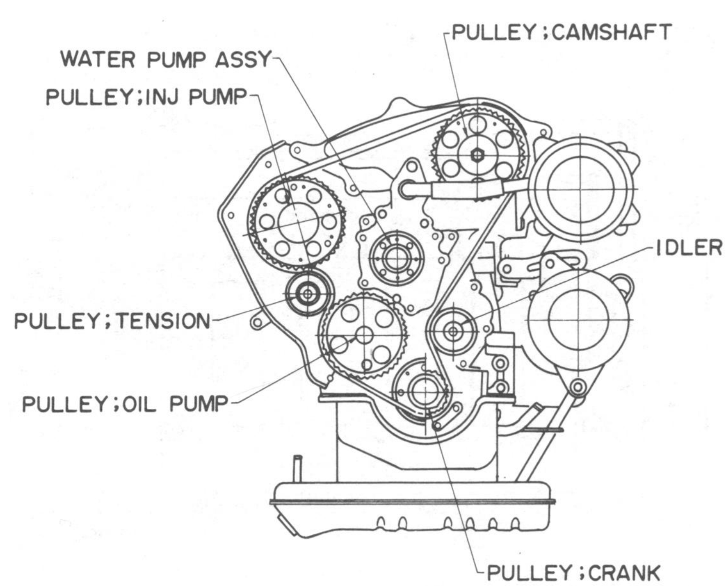


Fig. 18 - Timing belt layout

pickup is possible, as required. Injection nozzle is of the throttle type.

Fuel filter functions so that it can separate the water from the fuel. The water contained in the fuel is almost completely removed, and, at the same time, the dirt contained in the fuel is also removed, preventing the injection pump and nozzles from clogging. Reed switch is provided to warn the driver of an excessive amount of water accumulated. Fuel filter is of the cartridge type. Its maintenance service becomes very easy, as a result.

COOLING SYSTEM - Water pump is of the centrifugal type. Thermostat is of bottom bypass type with a jiggle valve equipped. Cooling system is shown in Fig. 19.

Cooling fan has seven blades made of polypropylene. Its outside diameter is 390mm. A fluid clutch of the temperature sensing type is incorporated. At low temperatures, the fan runs idle, contributing to an improved engine warm-up, fuel saving and noise reduction.

LUBRICATING SYSTEM - With the front wall of the cylinder block, the trochoid rotor housing of the oil pump is built in. The pump is driven by the timing belt pulley. Oil filter is of the full-flow cartridge type. Relief valve is mounted at the oil gallery portion of cylinder block. Bypassed oil is forced out through the oil jets, cooling the pistons. Oil cooler is of 6-step water-cooling plate fin type. This type of oil cooler prevents the lubricating performance from degradation caused by the raise of oil temperature at the time of sustained vehicle running at high speeds with full-load. Engine lubricating system is shown in Fig. 20.

ELECTRICAL EQUIPMENT - Starter is a lightweight and compact 2KW reduction type. With adequate gear ratio selected, excellent engine startability is secured. The alternator has a built-in IC regulator and a small vacuum pump Its capacity is 12V-50A. The Isuzu quick start-silent idling system (the Isuzu QSSI System-Fig. 21) is mounted in order to shorten the time required for glow plug pre-heating, to improve engine cold startability and to reduce engine and white smoke.

SPECIFICATION DETERMINATION OF COMBUSTION CHAMBER AND FUEL INJECTION SYSTEM

So as to secure a sufficient level of performance of diesel engines under any given environmental conditions, it is necessary for us to fully consider, at the stage of initial design, several specific conditions and engineer them into the specifications and design of the combustion chamber and fuel injection system.

The following three must be completly studied and ascertained at the time of engineering design work:

- Required Performances (covering engine output, fuel economy, exhaust emissions and smoke, etc.)
- Environmental conditions (covering fuel conditions on the market, climate and geographical conditions, etc.)
- Design specifications (covering engine compression ratio, fuel injection timing, etc.)

Table 4 shows the items used in our study for determination of the specifications of the combustion chamber and fuel injection system of the 4FB1 1.8-liter diesel engine. Their specifications can best be determined by having the required performances shown on the left of this table well balanced with each other. These requirements, however, are greatly affected by the environmental conditions shown to the right specified by 'o' mark. Engineering design work has to be made with these specific environmental conditions fully taken into consideration, which is adequately supported by required confirmation testing. The specifications to be determined are shown to

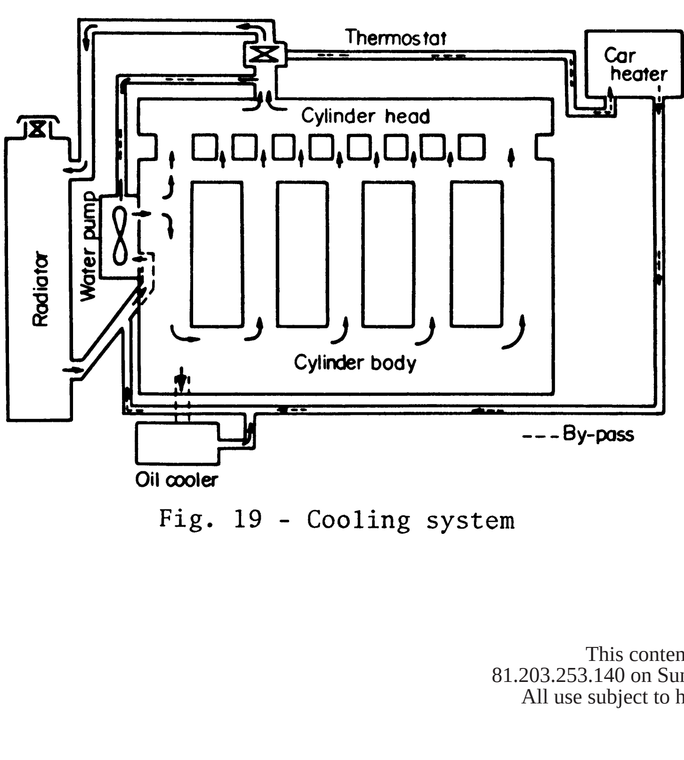


Fig. 19 - Cooling system

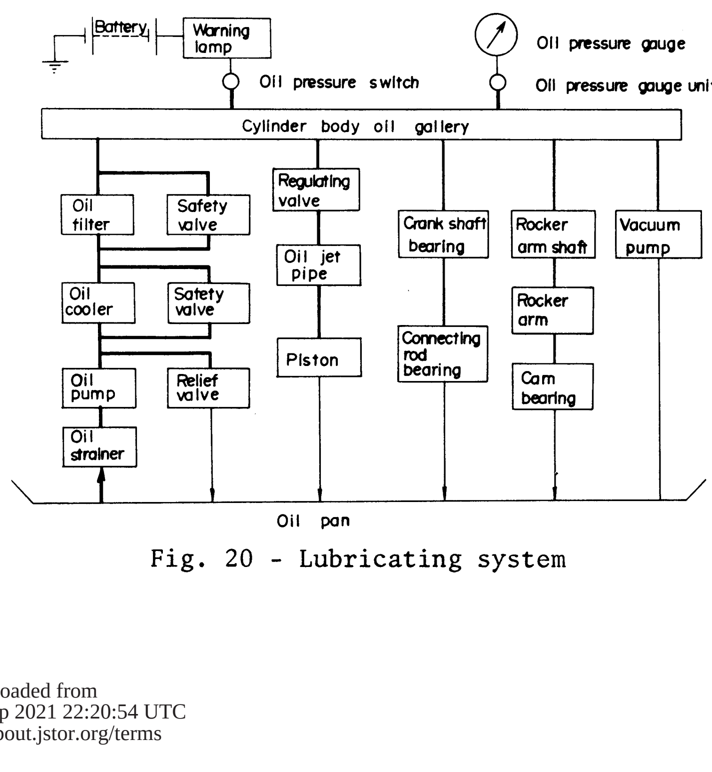


Fig. 20 - Lubricating system

the extreme right of this table. Those specified by 'o' mark greatly affect the required performances shown to the left on the same horizontal lines. Specification determination sequence was decided according to the priority given to these required performances.

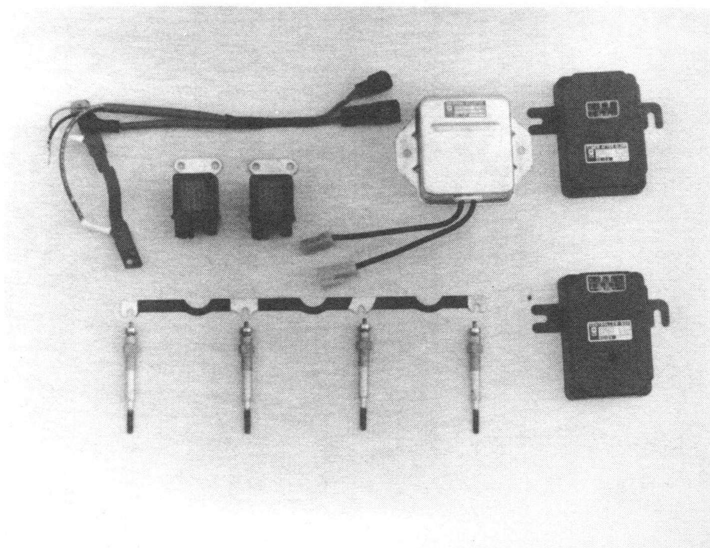


Fig. 21 - QSSI system parts

In the case of the 4FB1 1.8-liter diesel engine, the first priority was given to the fuel economy. Diesel-powered passenger cars have been blamed for their excessive noise and vibration so far. Despite these rather inherent drawbacks, diesel-powered passenger cars have favorably been accepted by a certain segment of the customers because of their good fuel economy. Our efforts were directed to a further upgrading of fuel economy of this engine for enhancing the image of fuel-efficient diesel-powered passenger cars.

As shown in Table 4, we have two parameters which contribute to the improvement of fuel economy. One is the engine compression ratio, and the other is the injection timing. From the fuel economy-standpoint, in the case of diesel engines, the lower compression ratio and advanced injection timing are preferable. These relations are summed up in Fig. 22. The specifications of engine should, therefore, be determined as: the compression ratio be kept low and the injection timing be set advanced. As shown in this figure, however, when the compression ratio is lowered, it will adversely affect the engine startability and exhaust HC emission and white smoke, and, likewise, advanced injection timing adversely

Table 4 - Relationships among required performances, environmental conditions and design specifications

REQUIRED PERFORMANCES	ENVIRONMENTAL CONDITIONS				DESIGN SPECIFICATIONS						
					CONBUSTION CHAMBER			FUEL INJECTION SYSTEM			
	FUEL CETANE NUMBER	HIGH ALTITUDE	COLD REGION	HOT REGION	COMPRESSION RATIO	THROAT AREA RATIO	SWIRL CHAMBER VOLUME RATIO	INJECTION PUMP PLUNGER DIAMETER	INJECTION NOZZLE	INJECTION TIMING	RESISTANCE OF FUEL LINE
FUEL ECONOMY					O					O	
ENGINE OUTPUT	O	O		O			O				O
EXHAUST EMISSION	O	O				O				O	
NOISE						O		O	O	O	
COLD STARTABILITY	O		O		O						
BLACK SMOKE		O									
WHITE SMOKE (MISFIRE)	O	O	O	O	O						O
INJECTION NOZZLE CLOGGING						O			O		

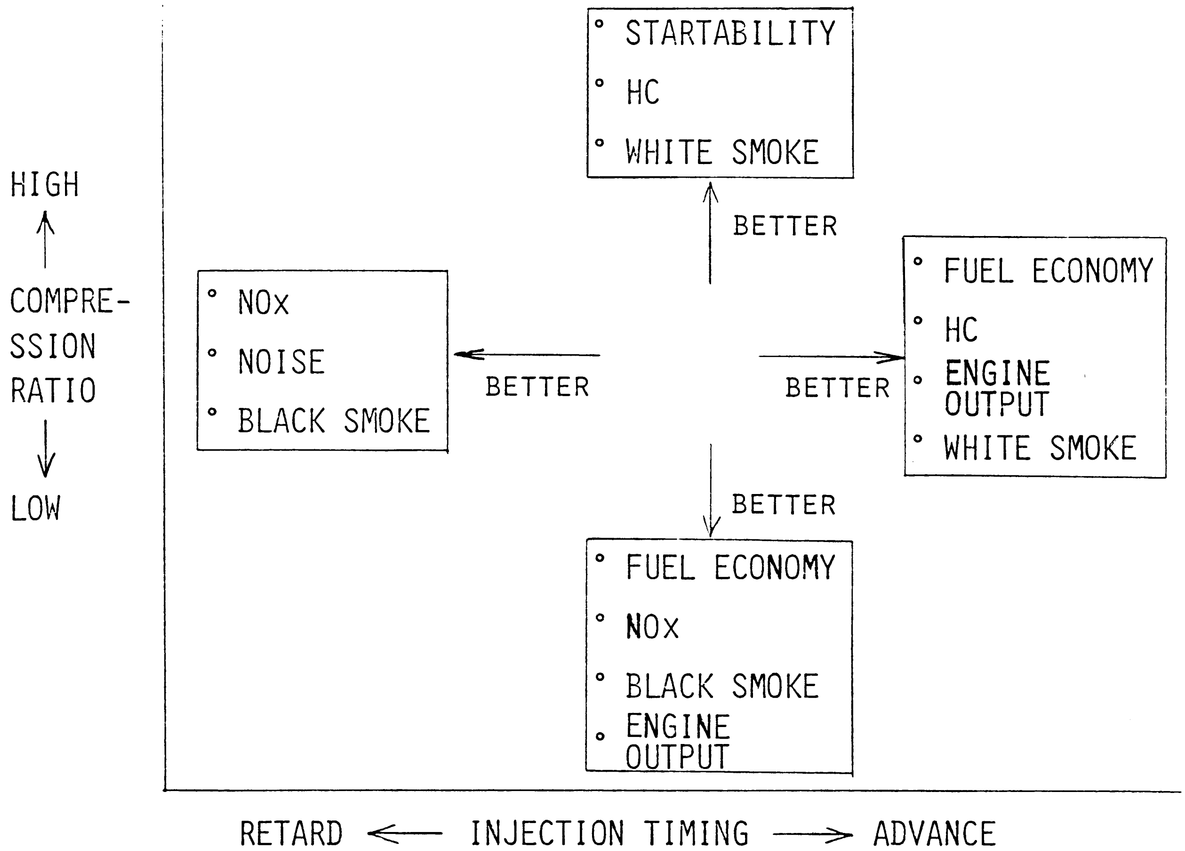


Fig. 22 - Required performances of compression ratio and injection timing

affects NOx emission and engine noise. We also had to pay attention to the tight NOx controls and the recent diesel fuel conditions in the United States. Fig. 23 shows an excerpt of the report on the trend of Cetane Number decrease - a part of DOE's Fuel Specification Trend Report. Apparently, Cetane Number of diesel fuels on the U. S. market shows a sharp decrease.

As shown in Table 4 earlier, the excessive lowering of the Cetane Number of diesel fuels has brought up several troubles in several aspects of engine performance such as engine misfire, HC emission increase, white smoke increase, etc. We had to engineer the combustion chamber and fuel injection system to be fully able to fight these troubles. Fig. 24 shows that the lowering of the Cetane Number of diesel fuel allows HC and Particulate emissions to increase. Fig. 25 shows a case where the lowered Cetane Number of diesel fuel makes the engine startability degrade and the white smoke at engine start increase. As mentioned earlier, the advanced injection timing is the best way to solve these troubles, but, to our regrets, with the penalties on the side of exhaust NOx emission and engine noises.

In the case of the 4FB1 diesel, another approach of a different kind was employed. In this approach the engine compression ratio was not set so high, but, with the use of additional devices, emission-related or other disadvantages were corrected. Engine compression ratio was set to 22. The ISUZU QSSI system with after-glow mechanism was adopted for better cold startability and running at idle. In addition, a block heater was mounted for the specific use of vehicles that are used at extreme cold temperatures below minus 25°C. Fig. 26 shows the results of our tests of the ISUZU QSSI system. Effectiveness of this system

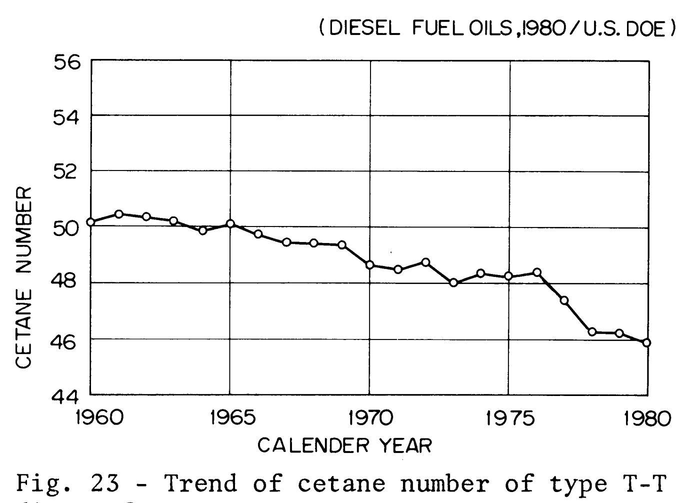


Fig. 23 - Trend of cetane number of type T-T diesel fuel

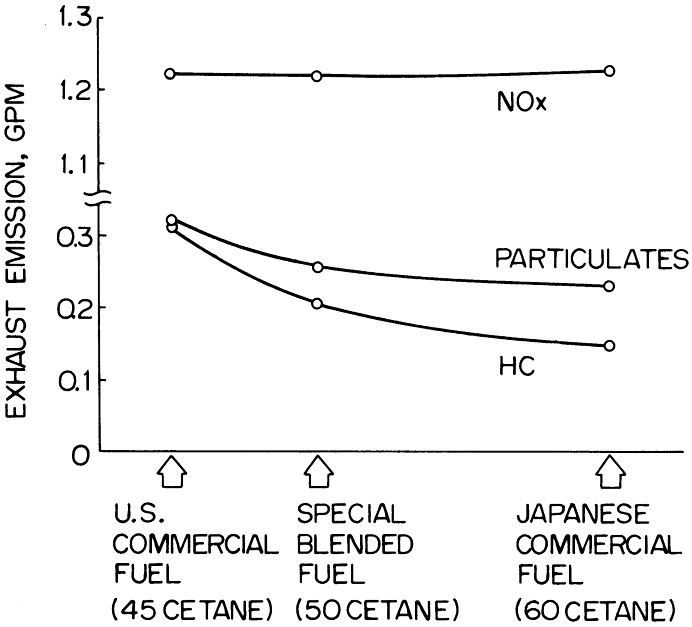


Fig. 24 - Effect of fuel cetane number on hydrocarbon and particulates

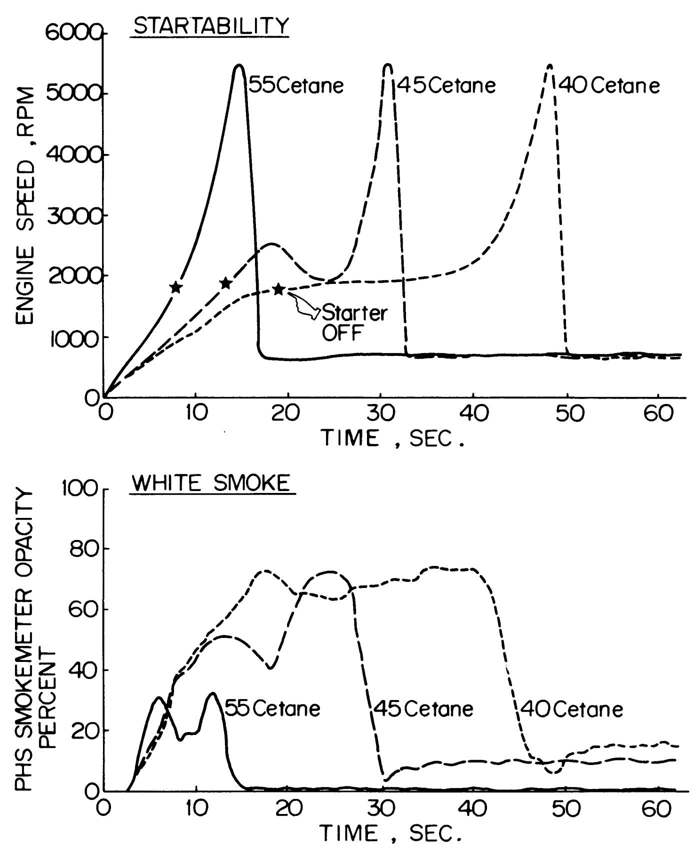


Fig. 25 - Effect of fuel cetane number on startability and white smoke

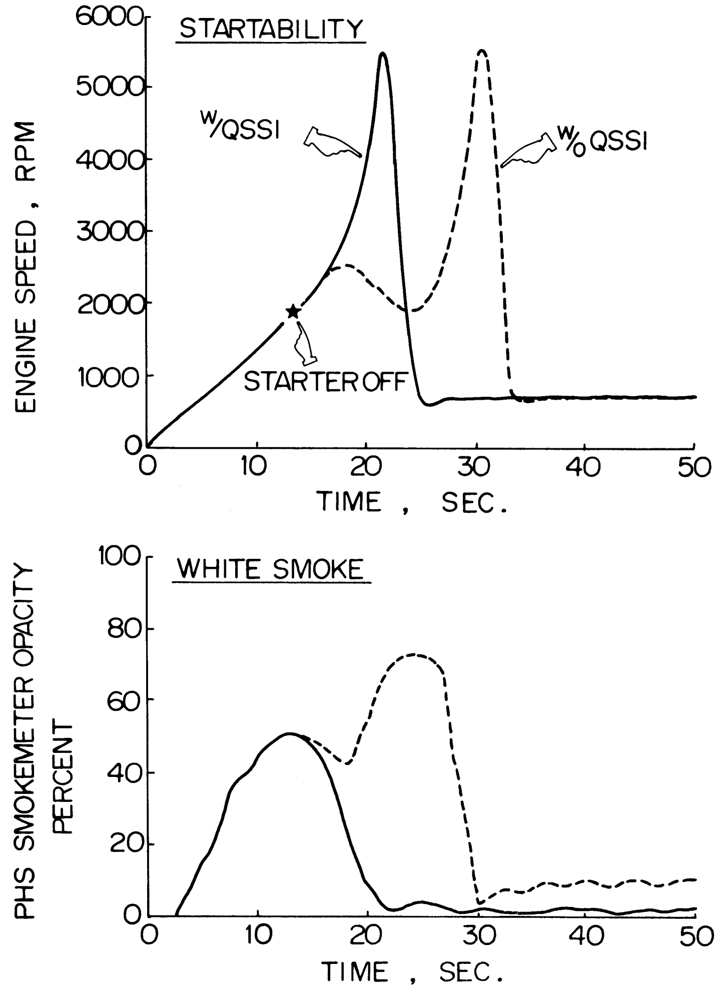


Fig. 26 - Effect of QSSI system of startability and white smoke

for an immediate engine start and a decreased exhausted white smoke is apparent.

After the engine compression ratio had been adequately determined, we went into the determination of the other specifications of combustion chamber. The first one we took up was the ratio of swirl chamber throat area to cylinder bore area (Area Ratio). Next was the ratio of swirl chamber volume to total combustion chamber volume (Volume Ratio). In determining these two ratios, we took notice of several performances required of the engine, that is, exhaust emissions and noise followed by fuel economy. Of these two, first, concerning exhaust emissions, we had to satisfy the U. S. statutory standards established. The trouble was that NOx control and HC control would conflict with each other.

The most influential parameters affecting the reduction of exhaust emissions and engine noise are, on the side of combustion chamber, Area Ratio and, on the side of fuel injection system, the diameter of injection pump plunger, the kind of injection nozzle and the injection timing. In these parameters, the injection

timing is most influential. Our check was made with several different settings of the injection timing so as to find the extent of their effect on the exhaust emissions and engine noise. The results are shown in Fig. 27. From the results we had obtained, we decided to have the injection timing set to the 'retarded' side with its penalty kept within the allowable limit. As shown in Fig. 28, the diameter of the injection pump plunger was determined to 8mm as that most preferable from the noise reduction standpoint. The injection nozzle having the smaller throttle area was selected. The variation of engine performance with different injection pump plunger diameters and injection nozzle throttle areas are shown in Fig. 29. The specifications of the fuel injection system thus determined did not favorably affect the engine output, however. We, therefore, tried to overcome this penalty by selecting adequate specifications of the combustion chamber.

Both Area Ratio and Volume Ratio can be important parameters affecting the engine output and the exhaust emissions. Effect of these two ratios on the engine performance is shown in Fig. 30. Area Ratio indicated on the 'X' axis in this diagram is closely related to the flow velocity from the main combustion chamber to the swirl chamber, which affects the swirl motion of the air-fuel mixture within the swirl chamber.
The bigger Area Ratio results in the weakened swirl motion which contributes to an increased combustion efficiency at high speed. Bigger Area Ratio also results in a decreased friction loss as well as an improved cold startability. When the swirl motion is made weaker, it makes lower the combustion efficiency and resultant engine output, and, at the same time, increases the exhaust smoke and noise during the range of low and medium engine speeds and decreases NOx emission and lowers the injection nozzle temperature which is attributable to the lowered combustion temperature.

Volume Ratio indicated on the 'Y' axis in this diagram is closely related to the gas velocity from the swirl chamber back to the main combustion chamber after the ignition and subsequent combustion. The bigger the Volume Ratio, the higher the gas velocity from the swirl chamber to the main combustion chamber. Gas velocity increase accelerates the air motion within the main combustion chamber, and, as a result, the engine output is increased and the generaation of the exhaust smoke is kept withheld. When the gas velocity is decreased by lowering the volume ratio, on the contrary, it favorably affects the fuel economy when the engine is running with part-load. These causal relations identified through our testing are shown in Figs. 31 and 32. The former is about the swirl chamber throat area, and the latter about the swirl chamber volume.

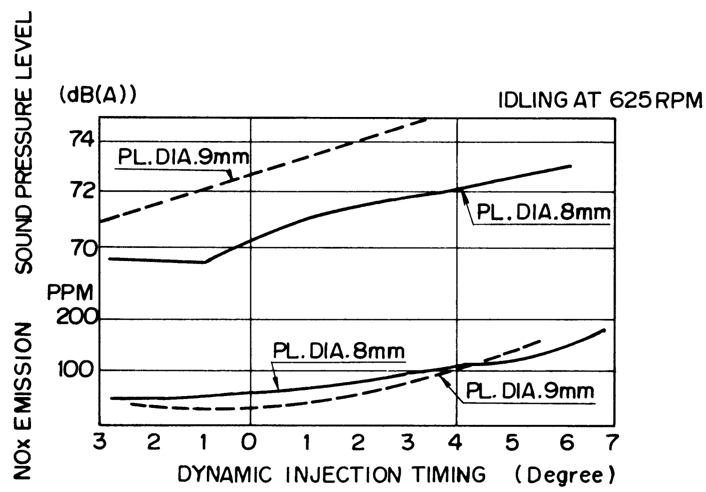


Fig. 28 - Effect of injection pump plunger diameter on engine idling noise and NOx emission

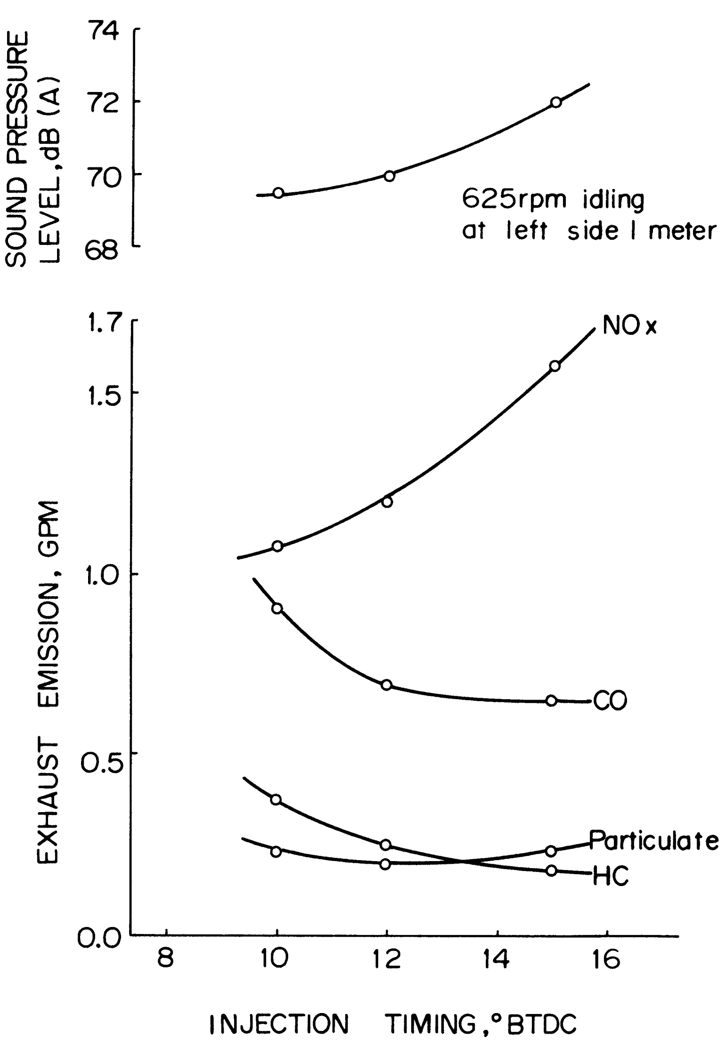


Fig. 27 - Effect of injection timing on exhaust emission and noise

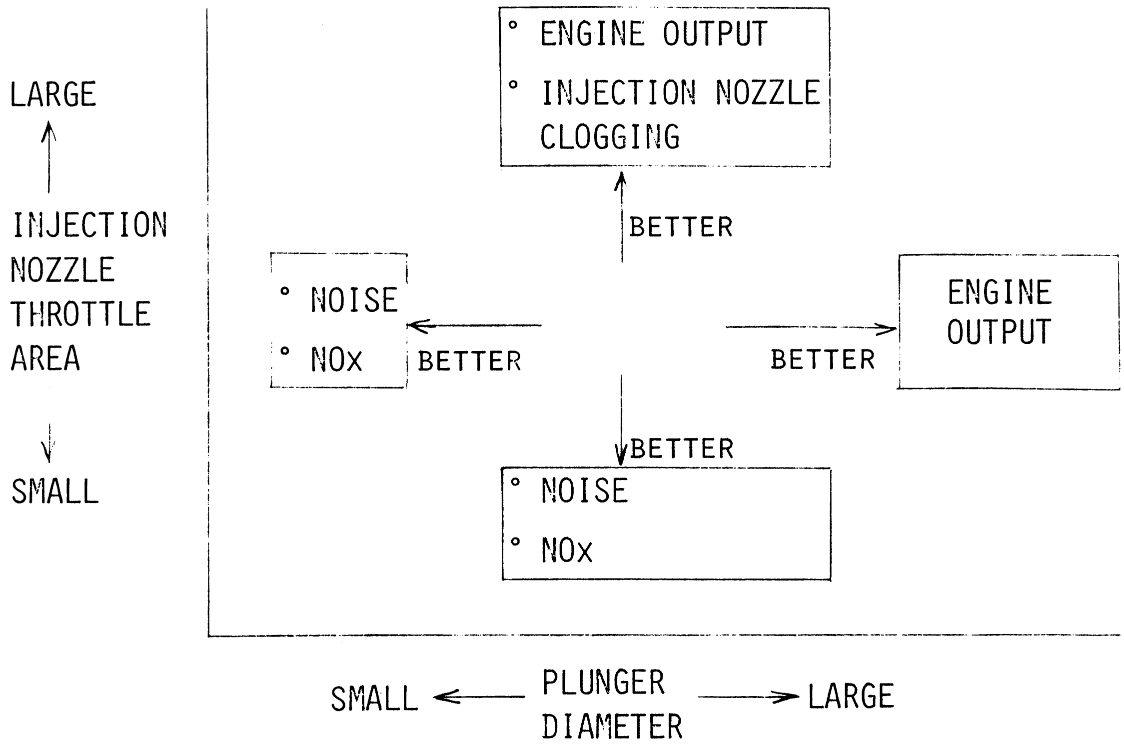


Fig. 29 - Required performance of plunger diameter and injection nozzle throttle area

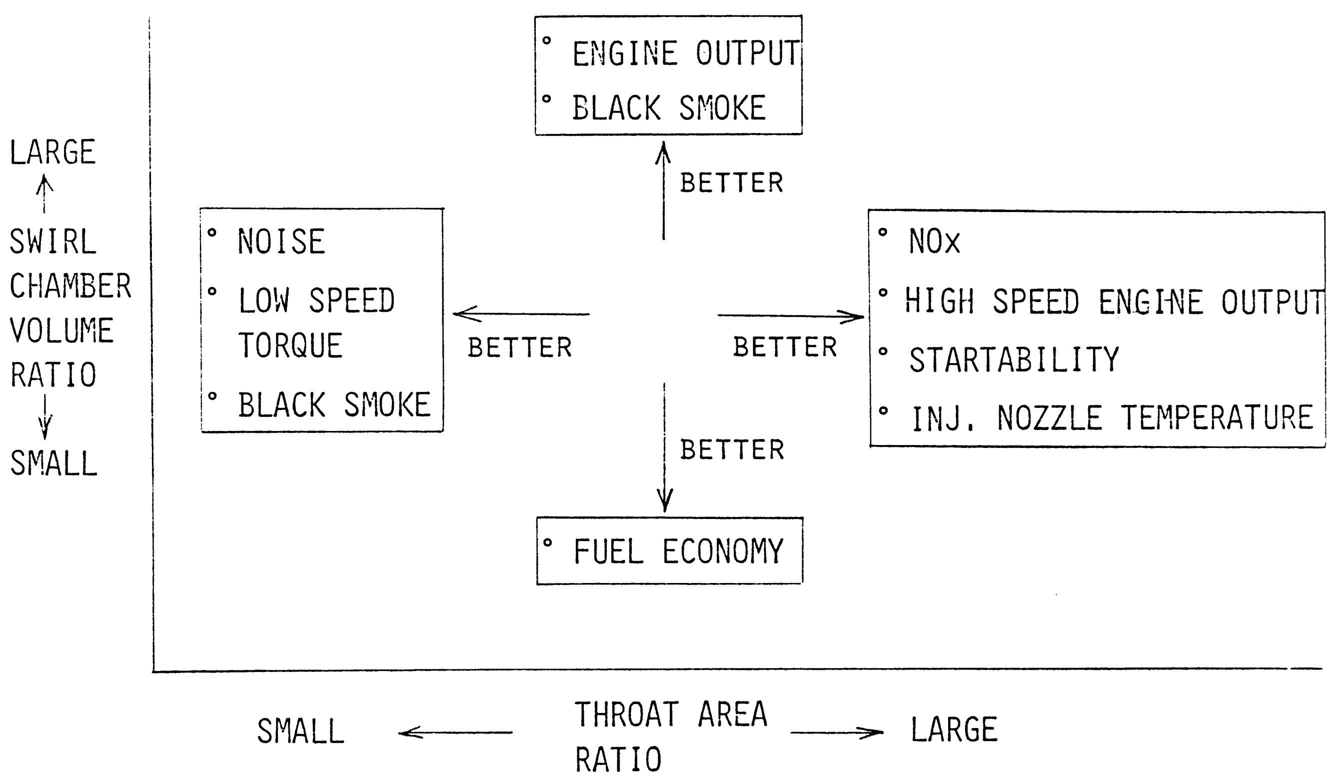


Fig. 30 - Required performance of area ratio and volume ratio

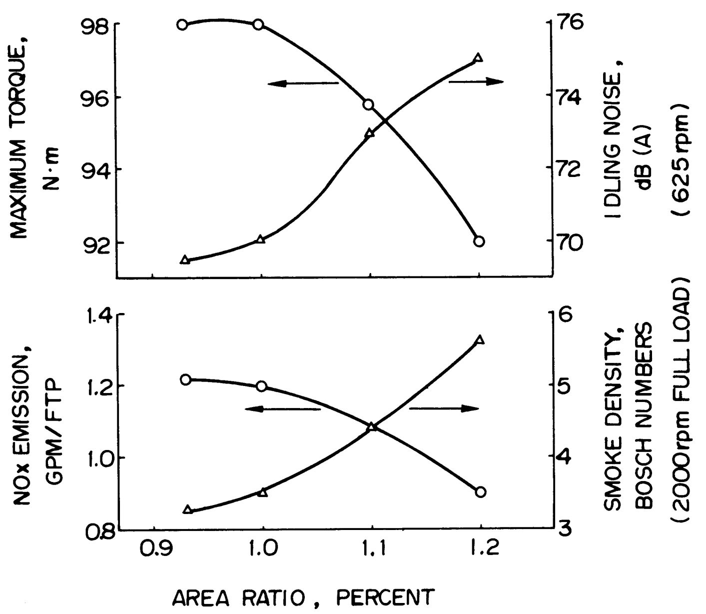


Fig. 31 - Effect of area ratio on engine torque, idling noise, NOx emission and smoke

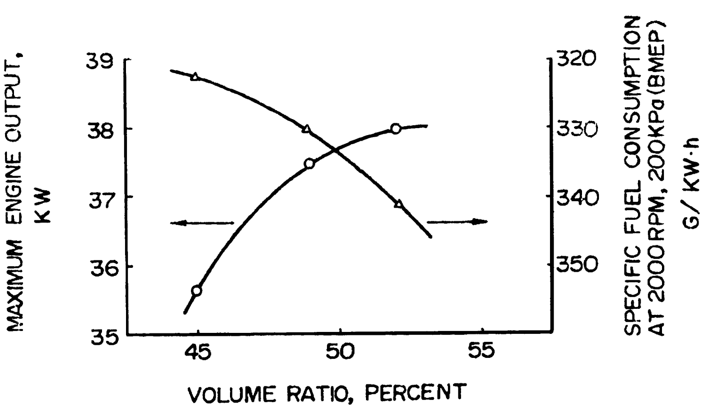


Fig. 32 - Effect of volume ratio on engine output and fuel economy

From the results of our study mentioned above, the Volume Ratio and the Area Ratio of this engine were determined to 49% and 1.0% respectively. By doing so, a satisfactory level of engine output has successfully been secured even under tight restrictions set for NOx emission and noise.
In addition to the test results mentioned above, the bigger Area Ratio contributes to the prevention of injection nozzle clogging.

FUEL ECONOMY

As compared to gasoline engines, diesel engines have a drawback - a relatively bigger friction loss with the vehicle running at high speeds. Despite this disadvantage, there also are several advantages in terms of fuel economy. They are the higher compression ratio, bigger air fuel ratio and less pumping loss when part load is given. As shown in Figs. 33 and 34, the 4FB1 1.8-liter diesel engine is several steps ahead of its gasoline counterpart of the same displacement in terms of fuel economy when running at low speeds and light load. Diesel-powered passenger cars mainly run with relatively light load and at medium engine speeds with stopping at idle. The running at over 4,000 rpm is seldom. From this, as shown in Fig. 35, diesel-powered passenger cars are from 30 to 50% better than their gasoline-powered counterparts in fuel economy.

For further upgrading of this engine in fuel economy, our engineering efforts had been concentrated on the following three areas:

1. Reduction of engine friction loss through the weight reduction of the engine moving parts, the decrease in the areas of the bearing contact surface, the adoption of optimum specifications for piston rings and the increase in cooling fan working efficiency.

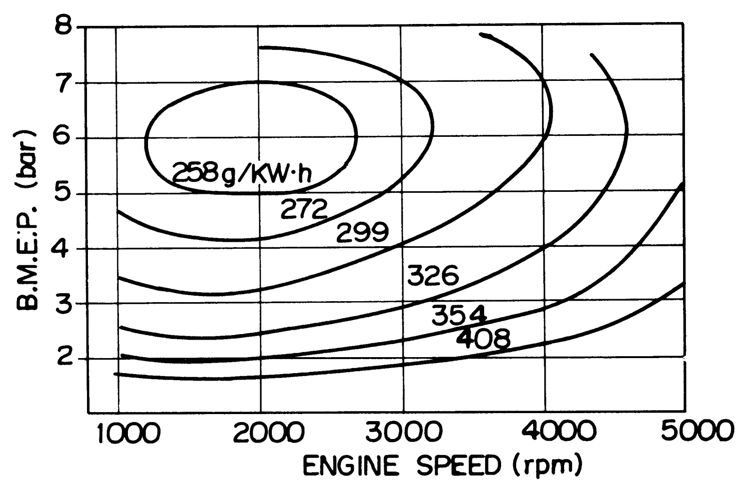


Fig. 33 - Fuel consumption map of 4FB1 diesel

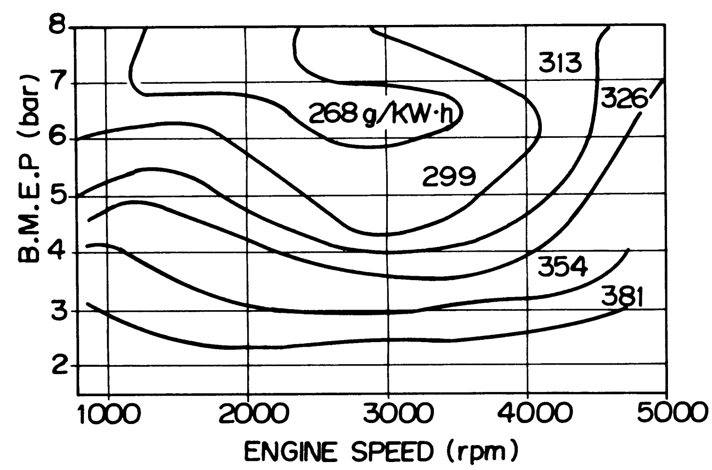


Fig. 34 - Fuel consumption map of G180Z gasoline

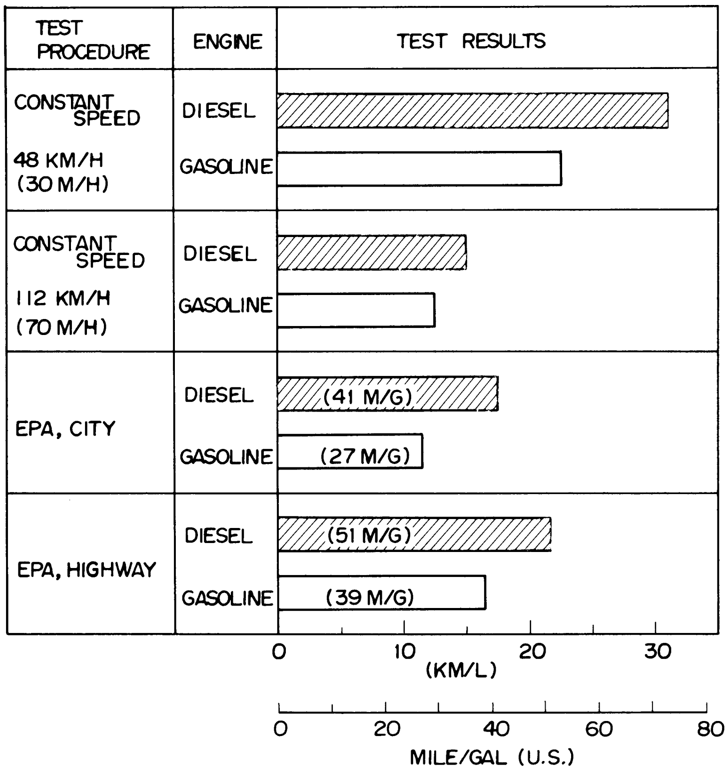


Fig. 35 - Fuel economy of Isuzu I-Mark, with 1.8L diesel or 1.8L gasoline

2. Improvement of fuel combustion efficiency through the adoption of the optimum combustion chamber configuration, the optimum compression ratio, fuel injection timing and fuel injection rate.
3. Best matching with the specifications on vehicle side through the determination of the engine output torque characteristics and transmission gear ratios.

Our explanation will go into Item 1 and 3 because Item 2 has been mentioned earlier.

REDUCED ENGINE FRICTION LOSS AND WEIGHT - As a method to reduce the engine friction loss, the basic structural dimensions of the 4FB1 diesel engine were finalized as close as possible to those of its gasoline counterpart as shown in Table 5. Dimensions of the crankshaft, connecting rod and piston were designed almost identical with those already adopted in its gasoline counterpart. The friction loss of the crankshaft of this engine was reduced below the level of the Isuzu conventional C190 1.95 liter diesel engine, as shown in Fig. 36. From this figure, the bearing dimension, Σdia^3 x length, corresponds, almost proportionally, with the friction loss.

As shown in Fig. 37, overall engine friction loss is decreased by reducing the weight of moving parts as well as by reducing the dimensions as shown in Table 6. This can favorably be compared with conventional C190

Table 5 - Major dimensions comparison, diesel engine vs. gasoline engine of Isuzu I-Mark

		Diesel 4FB1	Gasoline G180Z
Bore x Stroke	mm	84 x 82	84 x 82
Cyl. Block Height	mm	218.5	211.5
Conn. Rod Length	mm	133.5	133.5
Piston Comp. Height	mm	44.5	37.0
Crankshaft			
Main Journal Dia.	mm	56.0	56.0
Pin Journal Dia.	mm	49.0	49.0
Piston Pin Dia.	mm	25.0	22.0

Engine	Main Journal Dia	Bearing Width	Projection Area
4FB1	56 mm	23mm	6440mm²
C190	60mm	Front & Rear 31, (Center 38, Inter 23)	8760mm²

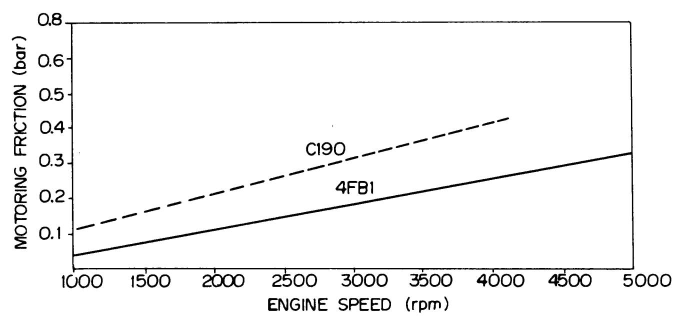


Fig. 36 - Crankshaft motoring friction loss

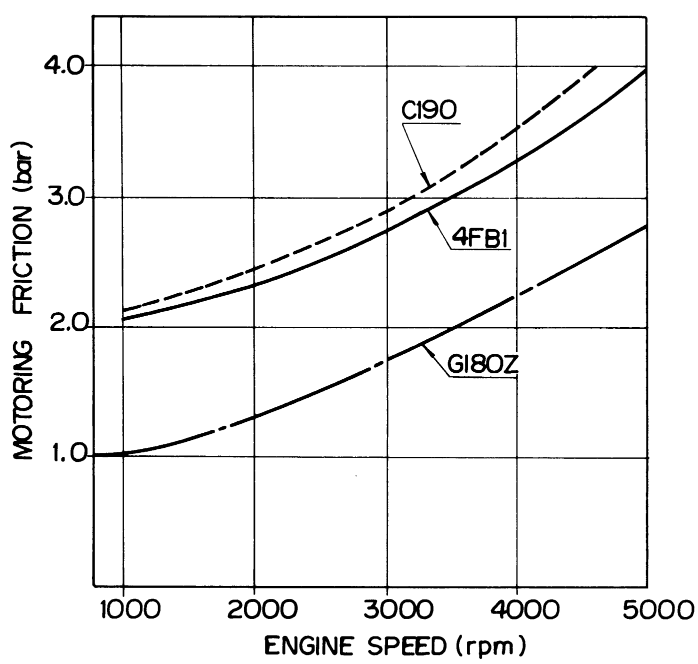


Fig. 37 - Motoring friction comparison - 4FB1, C190 and G180Z

Table 6 - Weight comparison of engine moving parts

Parts \ Engine Model	4FB1 (Diesel)	C190 (Diesel)	G180Z (Gasoline)
Piston (gr)	537	675	360
Piston Ring (gr)	40	55	37
Piston Pin (gr)	195	225	100
Conn. Rod (gr)	705	1,115	630
Crank Shaft (gr)	14,700	18,800	15,700

diesel engine but still remains on the level 40% higher than that of gasoline counterpart.

The weight of the 4FB1 1.8-liter diesel engine is still 23% heavier than its gasoline counterpart of the same displacement, whereas, when compared to the conventional 1.95-liter diesel engines, it is 20% lighter, as shown in Fig. 38. In this comparison, the weight of the gaoline engine includes the weight of emission-related parts such as the EGR system components and the air injection reactor system. In a component-to-component comparison, the cylinder head of the diesel engine is considerably heavier than that of the gasoline engine. This is because the former is made of cast iron, in the case of this engine, whereas the latter is made of aluminum.

ENGINE-TO-VEHICLE SPECIFICATION MATCHING - The engine torque at low range engine speeds ranging from 1,000 rpm to 1,500 rpm, was increased through appropriate matching with the fuel injection system. By doing so, even when the final reduction gear ratio is low, good fuel economy can be obtained with less penalty on the side of the vehicle driveability. As shown in Fig. 39, by the adoption of the high torque on the engine side and by the optimum selection of transmission gear ratio and final reduction gear ratio on the vehicle side, vehicle driveability close to that of gasoline-powered vehicles has been obtained as well as the fuel economy improvement.

HIGH PERFORMANCE

For obtaining an upgraded drive performance of diesel-powered vehicles comparable to gasoline-powered vehicles, it is necessary to use high-speed engines (in the case of the 4FB1 1.8-liter diesel, the rated speed of 5,000 rpm), and have the weight of engine components reduced for obtaining good fuel economy and reducing the friction loss of the moving parts.

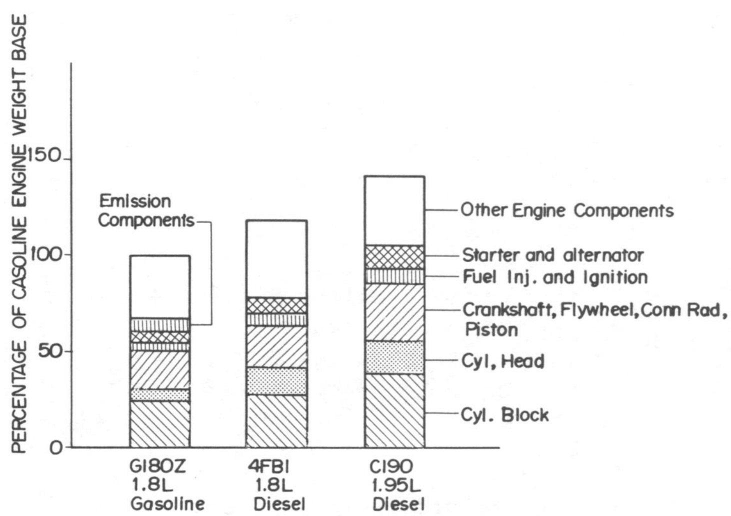


Fig. 38 - Weight comparison of engine components

Following counteractions were taken to build a high-speed and light-weight engine.

ACTUAL STRESS ON CRANKSHAFT - The 4FB1 1.8-liter diesel engine shares the basic dimensions of crankshaft with its gasoline counterpart, but, has a flywheel bigger than that is mounted on its gasoline counterpart. By adoption of the bigger flywheel, however, the bending stress on the crankshaft became excessive when the engine was running in the overrun range exceeding the rated engine speed. This excessive bending stress was reduced by stiffening the cylinder block and bearing cap and securing an adequate bearing clearance as well as an adequate matching of the crank pulley damper. Bending vibration occurs along with the torsional vibration. The bending stress was reduced by damper matching as shown in Fig. 40.

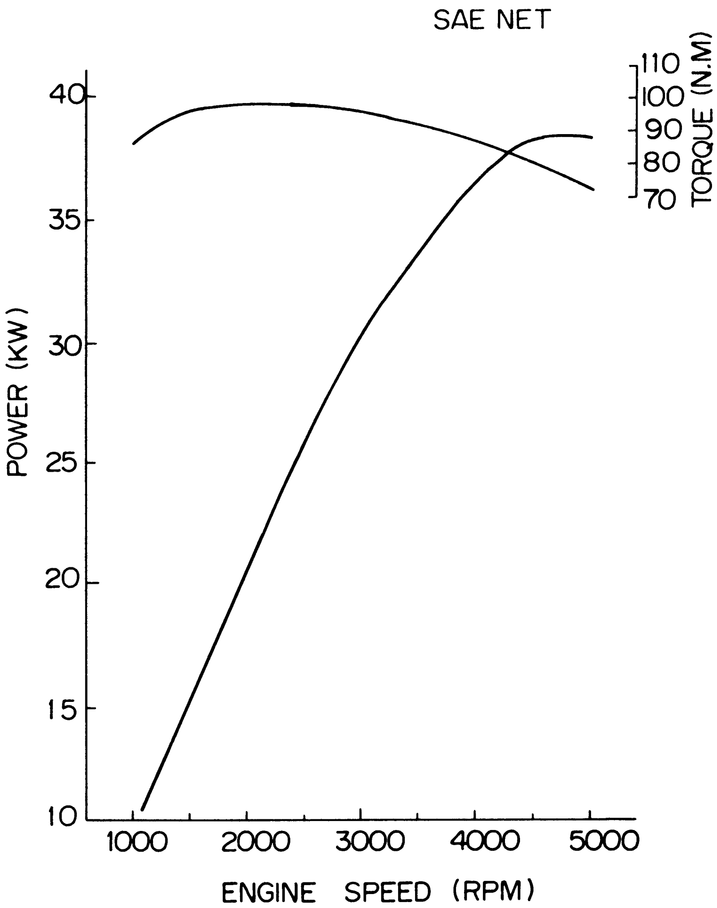


Fig. 39 - Performance of 4FB1 diesel engine

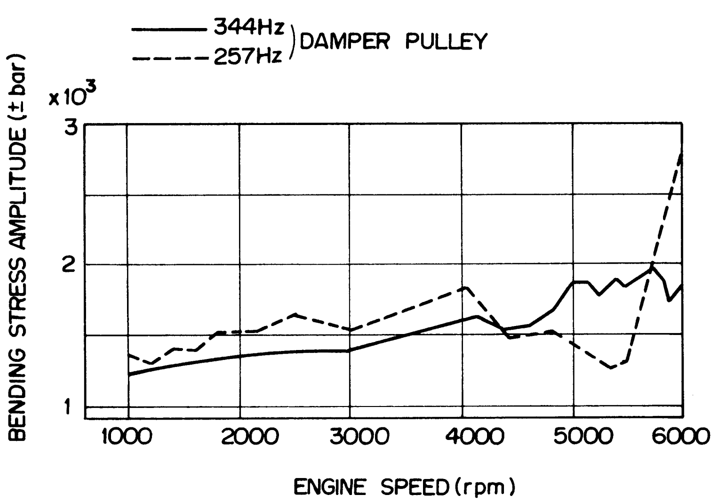


Fig. 40 - Crankshaft #4 pin bending stress

THERMAL LOAD ON PISTON - Fig. 41 shows the relation between the engine displacement and the amount of fuel supplied per liter at the maximum engine output. Plotted in this figure are the values obtained from most diesel engines below 3 liters in displacement now on the Japanese market. The values on the vertical axis represent the thermal loads on the engine within a given unit time. From this figure, we found that the 4FB1 1.8-liter diesel engine had an exceptionally high thermal load. Fig. 42 shows the temperature distribution measured at the point 3mm above the top land of the piston. From this measurement, we found that the temperatures were high and that there existed a big variation of temperatures between four cylinders. The results are shown by the solid lines in this figure. For reducing this high temperature, two methods were used. One was to secure an adequate flow of cooling water within the cylinder block. The other was to ensure adequate piston cooling. For securing an adequate flow of cooling water, the water duct was provided as shown in Fig. 43. As we can find from this figure, the flow of cooling water within the cylinder block was made uniform by adoption of the water

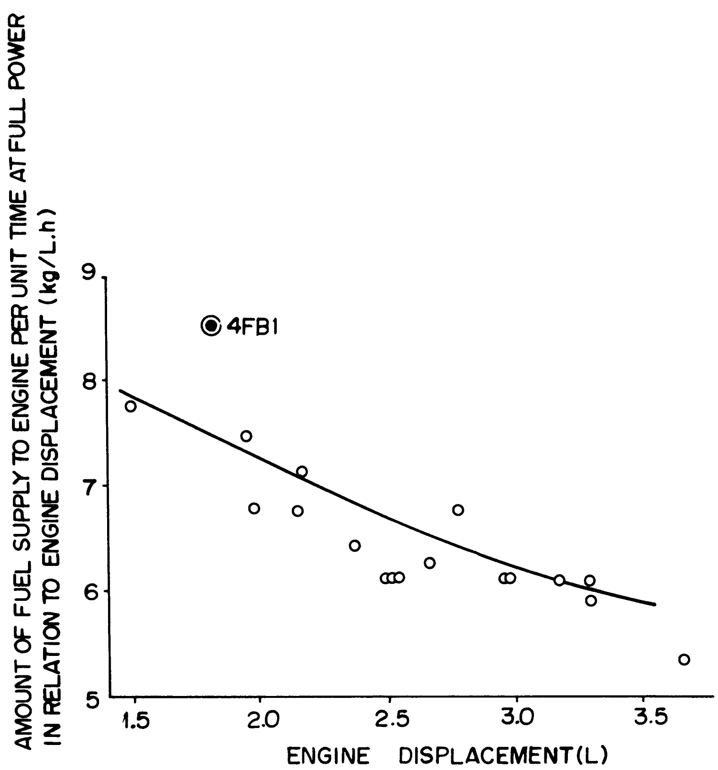


Fig. 41 - Comparison of thermal loading

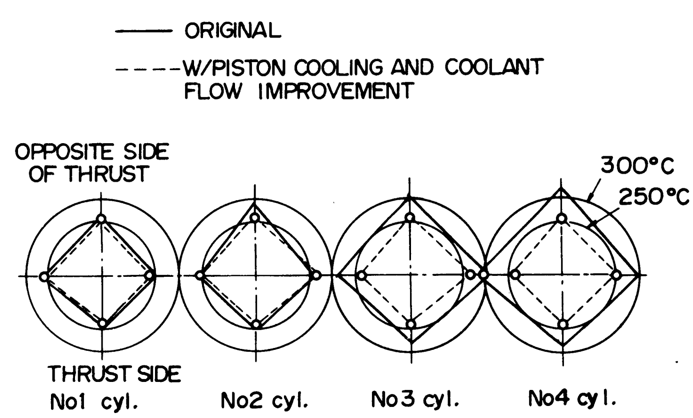


Fig. 42 - Distribution of piston temperature

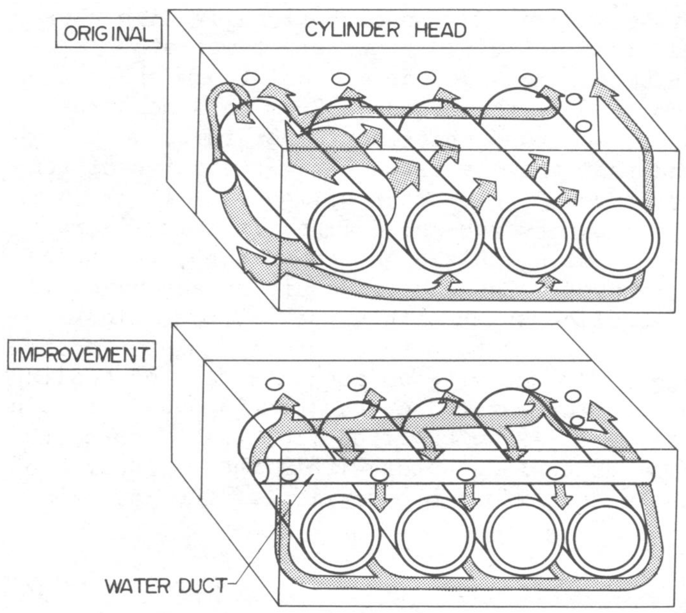


Fig. 43 - Improvement of coolant flow in cylinder block

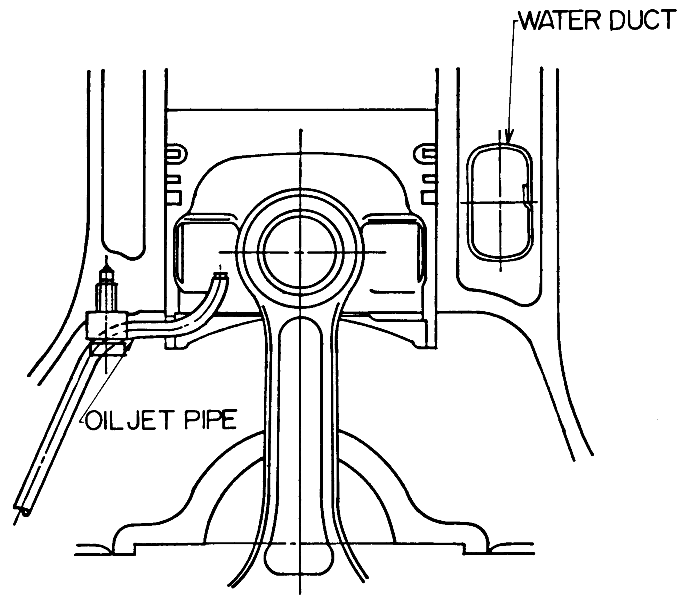


Fig. 44 - Piston cooling and water duct

duct. Speaking of the piston cooling and oil jet shown in Fig. 44, they were accomplished by utilizing the engine lubricant bypassed from the oil pressure regulating valve. By adoption of these two methods, the piston temperatures were reduced to the level shown in Fig. 42, by the dotted lines. As we can find from this figure, a uniform level of piston temperature as well as lowering the piston temperature was obtained through all four cylinders. Higher thermal load on the 4FB1 1.8-liter diesel engine was thus successfully withheld, and a high engine speed of 5,000 rpm was also successfully achieved.

MOUNTING OIL COOLER - In the United States, continued high-speed vehicle running on the highway is frequent. The temperature rise of the engine oil is easily expected when the vehicle runs for a long distance. The increased engine oil temperature is likely to cause the bearing metal and the pistons to stick and other troubles to occur which include an excessive deterioration of the engine oil, or a shortened service life of the engine. Fig. 45 shows the temperatures of the oil gallery and No.3 journal (the center bearing of the crankshaft) measured on an engine which was running. As shown in this figure, the temperature of the bearing was some 10°C reduced by the adoption of the oil cooler, giving, as a result, an extent of additional margin to the bearing wearability. Based on the results of this test, an oil cooler has been mounted on the 4FB1 diesel engine to be offered to the U. S. Market.

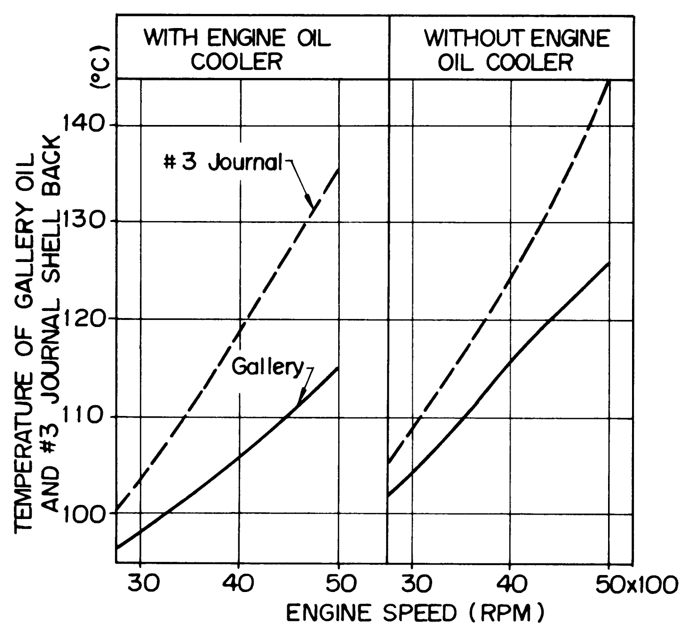


Fig. 45 - Effect of engine oil cooler on temperature at oil gallery and #3 journal bearing shell back

ENGINE NOISE REDUCTION

Diesel engine noise is high especially at idling. Several improvements had been made for the 4FB1 1.8-liter diesel engine. Engine noise of diesel-powered passenger cars are characterized by the following two types:

- Chattering Noise Coming Out of the Engine; The combustion gas pressure of diesel engines is higher than that of gasoline counterparts. This high combustion gas pressure causes the noises of engine knock and piston slap to develop.

- Rattling Noise Coming Out of the Drive Train; The fluctuation of the angular velocity of the crankshaft attributable to the within-cycle torque fluctuation causes the rattling noise to develop within the transmission and other drive-train components.

For the reduction of these noises, several counteractions were taken for the 4FB1 1.8-liter diesel engine as follows;

- Fuel Injection Rate was lowered. - for reduction of the engine knocking noise.
- Piston Clearance was made smaller. - for reduction of the piston slapping noise.
- Damper Pulley was mounted. - for reduction of the noise coming from the crankshaft.
- Flywheel Inertia Mass was increased. - for reduction of the rattling noise coming from the transmission.
- Fan Clutch was mounted. - for reduction of the fan noise
- Timing Belt was mounted. - for reduction of the noise coming from the timing chain or gears

For reduction of the engine knocking noise, the diameter of the plunger of the injection pump was determined to 8mm. By the use of this size plunger, the fuel injection rate was lowered, resulting in the reduction of the engine knocking noise.

For reduction of the piston slapping noise, the piston clearance was made smaller. In the new 4FB1 diesel engine, an autothermic piston has been adopted, as a result, the piston clearance has been made smaller, that is, reduced down to 25 μm. With the adoption of this type of piston, an appreciable improvement has been achieved in engine noise reduction, especially at lower engine speeds. At engine idling, a 3 dB(A) decrease has been achieved as shown in Fig. 46.

A rubber-made cogged belt has been mounted. On late model small diesel engines, this type of timing belt has extensively been used because of its excellent noise reducing property. Selection of the most appropriate type is necessary, however. Table 7 shows the evaluation factors we used in our test for selection of the belt most appropriate for the 4FB1 diesel. Table 8 shows the results of our comparative durability evaluation test made of three different kinds of rubber-made cogged belts. In the past, Uniroyal type 'C' tooth profile had been used most extensively. We, however, concluded Uniroyal type 'B' tooth profile, was most adequate for this engine from every aspect. As shown in Fig. 47, a sizeable amount of angular phase difference between the crankshaft and the injection pump was identified when the belt tension was changed in three ways, i.e., to 15Kg, 25Kg and 40Kg. From this test, we found that, when the belt tension was too weak, the angular phase difference was bigger. The reason is possibly that the fluctuation of the injection pump driving torque and the vibration of the timing belt itself resonate with each other, making the timing belt start vibrating. Based on the results of this test, the belt tension was determined to 25Kg. Reduced engine speed at engine idling contrib-

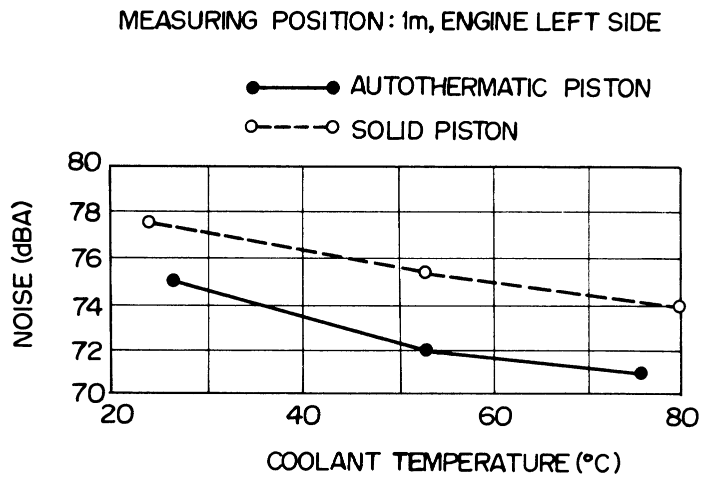


Fig. 46 - Effect of autothermic piston on engine idling noise

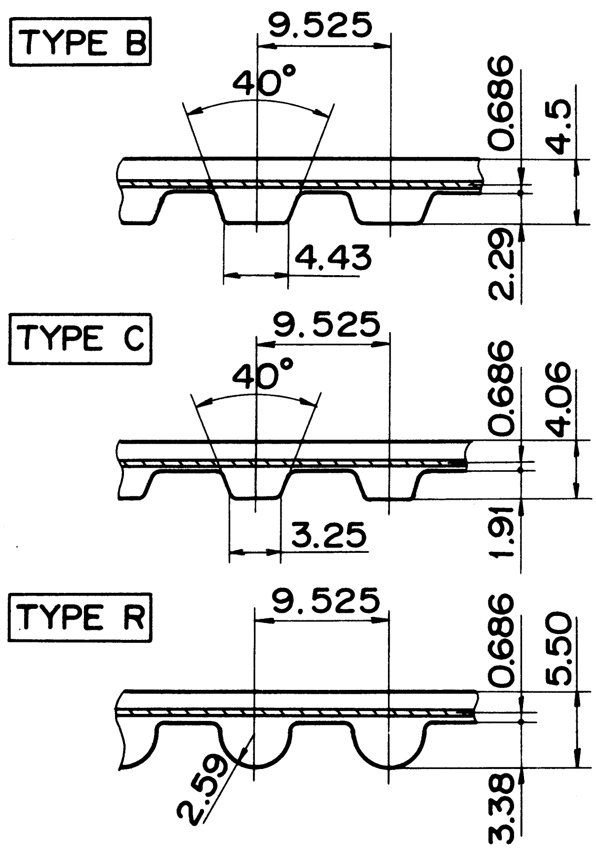
Table 7 - Evaluation of timing belt

SPECIFICATION OF TIMING BELT	NUMBER OF KINDS TESTED
1. TOOTH TYPE	3
2. CODE MATERIAL	2
3. RUBBER MATERIAL	4
4. COVER CLOTH MATERIAL	3
5. TOOTH WIDTH	3
6. BELT TENSION	4

MAIN EVALUATION ITEMS
1. TEMPERATURE RESISTANCE -40°C TO 120°C
2. DURABILITY
3. TOOTH SHEAR
4. JUMPING
5. WATER, OIL & DUST RESISTANCE
6. PHASE FLUCTUATION

Table 8 - Tooth type selection test results of timing belt

EVALUATION ITEM \ TOOTH TYPE	B	C	R
HEAT RESISTANCE	○	○	○
COLD RESISTANCE	○	○	○
DURABILITY	○	○	○
TOOTH SHEAR	◎	○	○
JUMPING	○	△	◎
WATER, OIL AND DUST RESISTANCE	○	○	○
MARKETABILITY	○	○	△



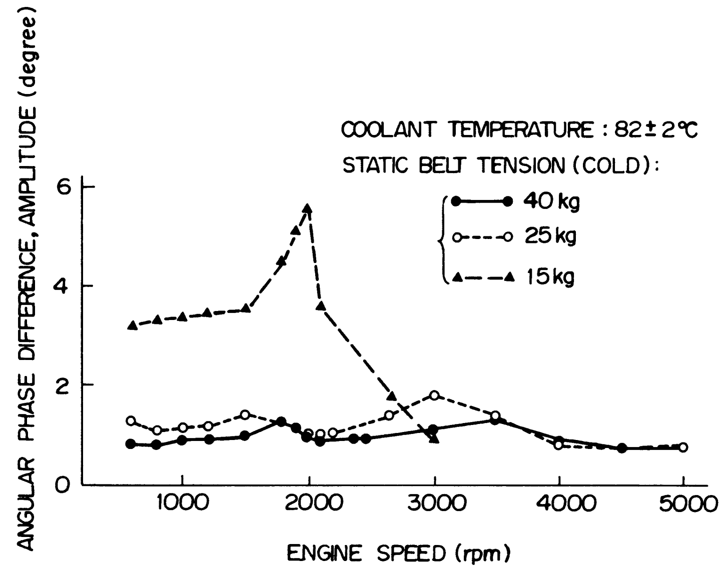


Fig. 47 - Influence of belt tension on angular phase difference between crankshaft and injection pump

utes to the reduction of the diesel engine noise. With the decrease in engine noise by reducing the engine speeds, the fluctuation of crankshaft angular velocity (or rotational movement fluctuation of the crankshaft within the cycle) increases in inverse proportion. Lowering the final reduction gear ratio is also another method to reduce the engine noise while the vehicle is running. This also contributes to the fuel economy improvement as mentioned earlier. This, however, means that the engine has to run at the lower engine speeds constantly. When the engine is running at lower speeds, the angular velocity fluctuation of the crankshaft is big, allowing several vibration and noise-related problems to come up associated with the torsional vibration of the vehicle drive-line. The extent of the angular velocity fluctuation of the crankshaft is closely related to the extent of the stress exerted on the teeth of the timing belt.

As the counteraction to that, the angular velocity fluctuation of the crankshaft at engine idling was set to 110 rpm by increasing the flywheel inertia mass and by minimizing the between-cylinder variation of the fuel injection volume at engine idling. The results are shown in Fig. 48. The engine speed at idling was reduced to the level of 625 rpm. The final reduction gear ratio was also lowered to 3.15 in the case of a 4-speed manual-transmission.

A kind of intermittent tapping noise was heard from the vehicle underside. This noise was caused by the vibration of the crankshaft due to an increase in the bending stress exerted thereon. We examined the extent of the vibration acceleration at the engine rear mounting portion, finding that the resonance occurred at the point of 250Hz. The resonance amplitude through all four cylinders was analyzed, and , as a result, we found that the vibration of the crankshaft occurred most severely after the combustion in the 4th cylinder occurred, as shown in Fig. 49. If the fuel injection into the 4th cylinder stopped, this vibration peak disappeared. From this fact, we assumed that the biggest bending stress was exerted on the crankshaft when it was at its 4th cylinder position. As the counteraction

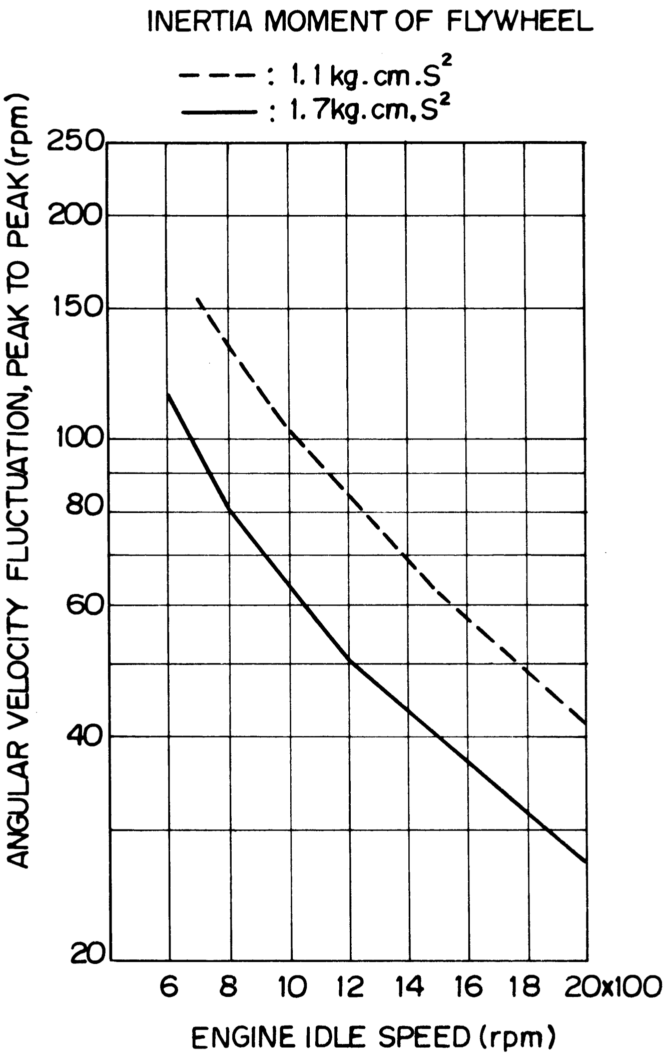


Fig. 48 - Influence of flywheel mass on angular velocity fluctuation

to that, the strength of the crankshaft was reinforced, especially at its 4th cylinder position. On the cylinder body side, reinforcement was also made on its corresponding portions. The results are shown in Fig. 50.
By the application of these counteractions, a tapping noise once heard from the vehicle underfloor mostly disappeared. Fig. 51 shows the noise level obtained through a microphone placed one meter apart from the left side of the engine. The test engine had been counteracted, as mentioned earlier, for noise reduction. The same extent of noise insulation and reduction measures were taken on the vehicle side, too. As shown in Figs. 52, 53 and 54, the vehicle noises, both interior and exterior, have been reduced down to the levels well comparable to those of gasoline-powered vehicles.

COUNTERACTIONS TAKEN FOR HIGH-ALTITUDE USE

In the mountain regions of the United States, high-altitude areas (over 1,500 meters above the sea-level, for example) widely expand. Highways running the districts as high as over 3,000 meters above the sea-level are not unusual.

In the case of the diesel engine, the fuel is injected into the engine, irrespective of the difference in the atmospheric pressure, that is, irrespective of the difference in the altitude. When the diesel engine runs at high altitude, its air-fuel ratio is reduced when the combustion takes place, which results in an inefficient combustion with resultant lower engine output when running at full load. Generation of an excessive amount of exhaust smoke and CO emission will be inevitable. Smoke is not only exhausted to the atmosphere but also forced into the engine oil. The smoke forced into the engine oil deteriorates

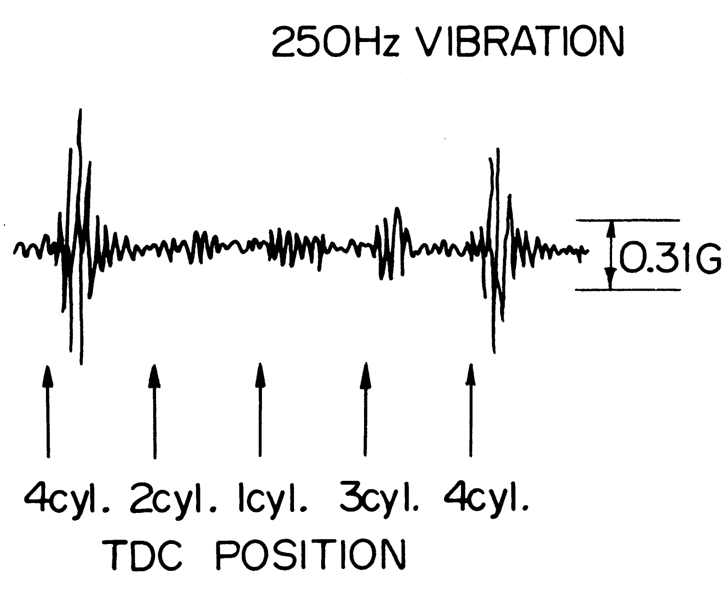


Fig. 49 - Vibratory acceleration of engine rear mount portion

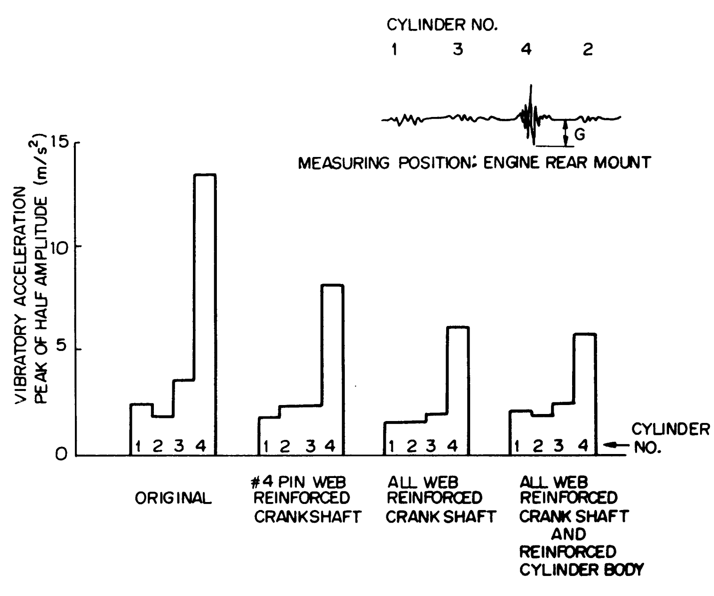


Fig. 50 - Influence of the reinforced parts on vibratory acceleration

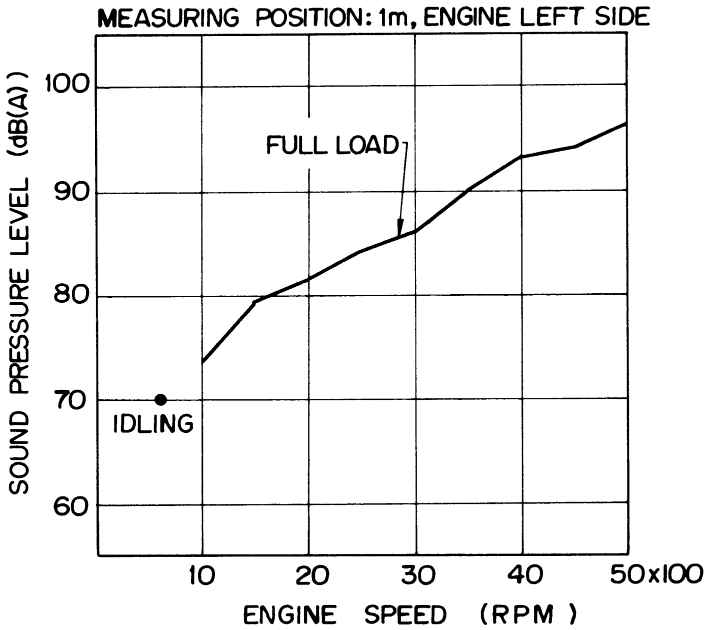


Fig. 51 - Noise of 4FB1 engine unit

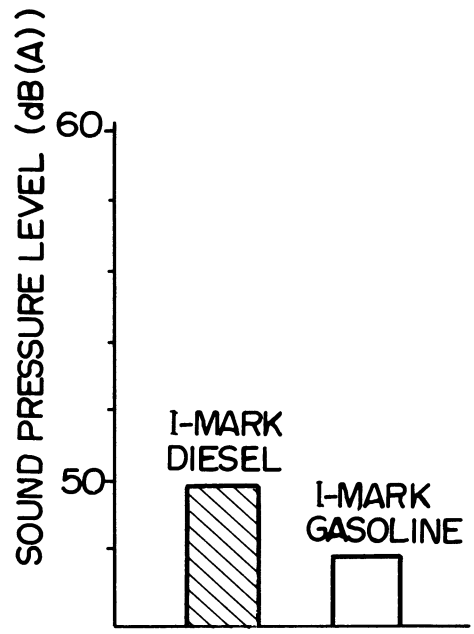


Fig. 53 - Interior noise of I-Mark diesel and gasoline while idling

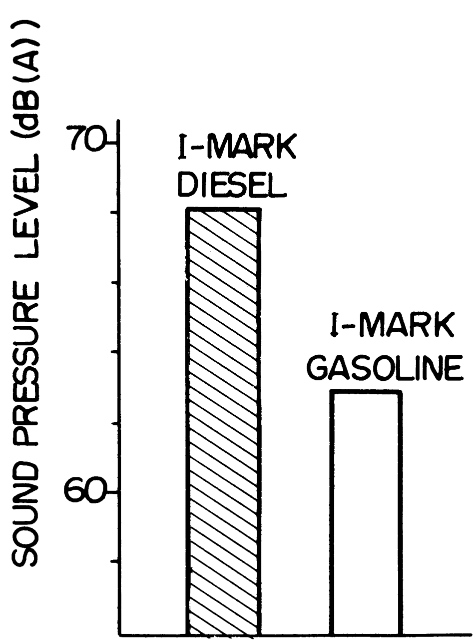


Fig. 52 - Exterior noise of I-Mark diesel and gasoline while idling

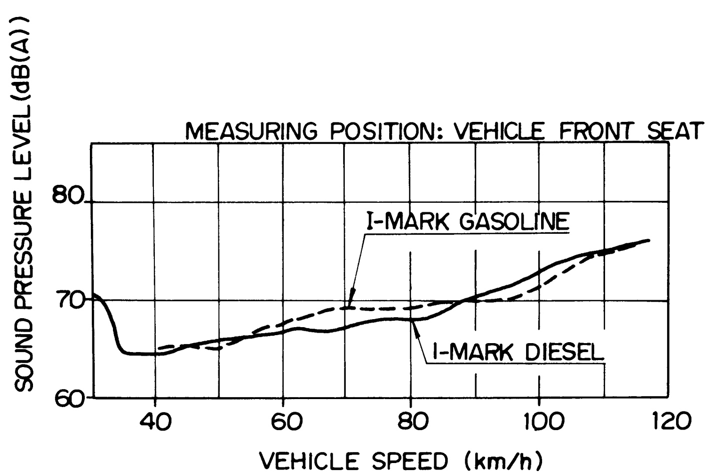


Fig. 54 - Interior noise of I-Mark diesel and gasoline

it when the engine runs for a sustained long time, affecting adversely the lubrication of the engine components. Exhaust smoke control at high-altitude is of utmost necessity.

In the Isuzu 4FB1 1.8-liter diesel engine, a high-altitude compensator is provided as shown in Fig. 55. Its mechanism is such that the maximum amount of fuel injection is regulated by the function of the bellows which expands or contracts according to the atmospheric pressure change. For best determining the specifications of this compensator, a special testing deivce was developed for reproducing the atmospheric conditions at high altitude within the laboratory. A surge tank was attached on the intake side of the engine. A butterfly valve was installed at the inlet of the surge tank. This valve worked to change the pressure within the tank in several ways as required. By using this testing device, the atmospheric pressure of the intake air which we experience at high-altitude could easily be simulated.

Fig. 56 shows the results of the engine performance and exhaust smoke simulated by using this device. As we can find from this figure, the engine output came down at high-altitude. This lowering tendency continues even when the fuel flow is increased. The reduction of the exhaust smoke occured even when the fuel flow remained small. In determining the specifications of the com-

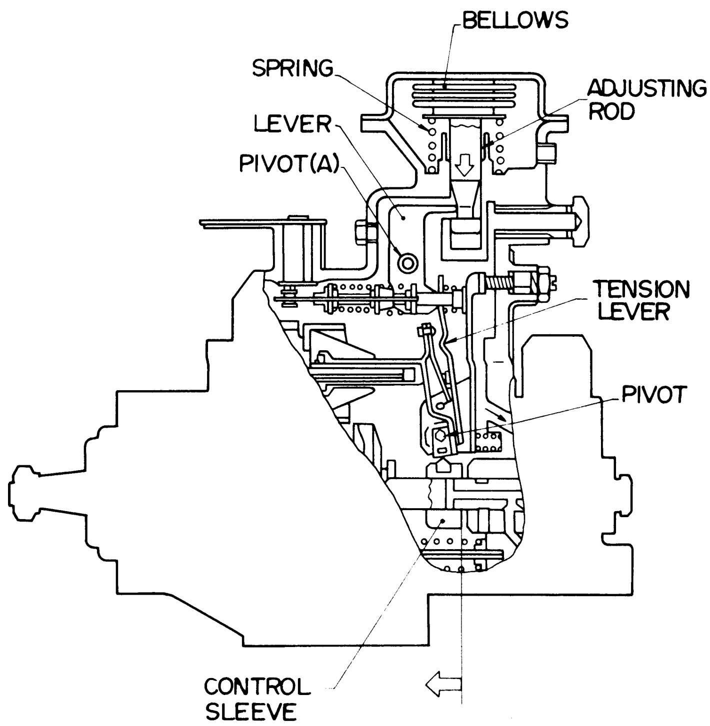


Fig. 55 - Aneroid type altitude compensator

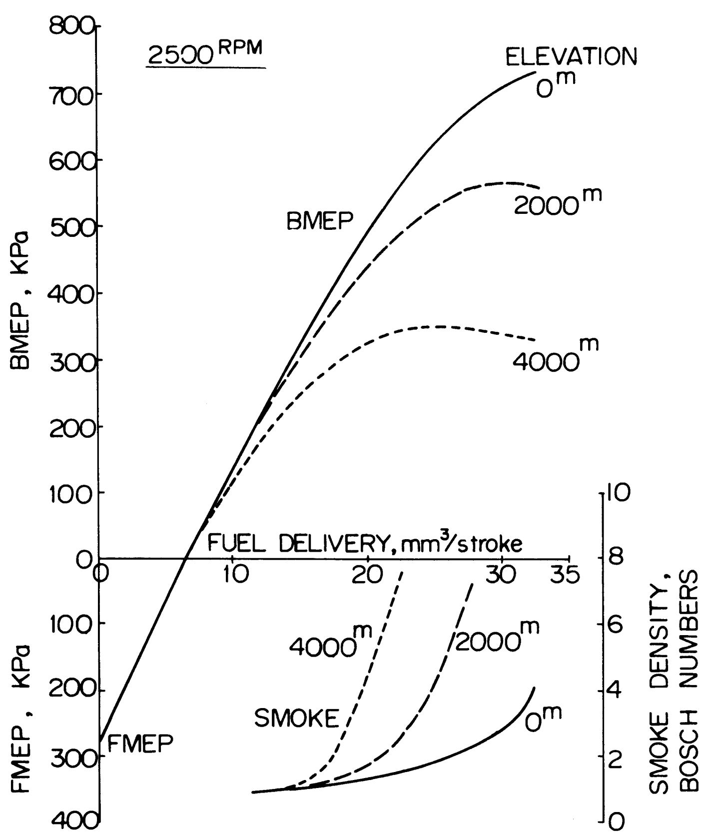


Fig. 56 - High altitude simulation test results on engine performance and exhaust smoke

pensator, the extent of the smoke density expected was weighed against the extent of the engine output desirable for the running of the vehicle and a reasonable compromise was made between them. For this determination, this figure was used fully. Fig. 57 shows the effect of the high-altitude aneroid compensator mounted on the 4FB1 diesel, on the exhaust smoke control and engine output. With the least penalty on the side of the engine output, the big increase in smoke development was adequately withheld.

Our engineering consideration was given to the fuel system, too. Diesel engine running at high-altitude has a number of troubles such as engine misfire, generation of excessive white smoke, and decrease in engine output. Lowered atmospheric pressure causes within-cylinder temperature to lower, inviting ignition difficulty. This lowered atmospheric pressure also allows air bubbles to develop within the fuel. These air bubbles are forced into the injection pump. This condition makes the fuel pressure drop within the injection pump, making the auto-timer go towards the retard side of the injection timing. This injection timing retard causes surge in the vehicle running at the wide-open-throttle acceleration. For prevention of these problems, the combustion chamber of the engine has to keep a certain level of compression ratio at least, and, at the same time, the resistance to the fuel flow in the fuel line has to be kept to the minimum for preventing the air bubbles from developing within the fuel. Engineering consideration had also to be given to the layout of the fuel line to prevent the air bubbles.

The engine compression ratio of 22 was decided as mentioned earlier. Our efforts were given to the designing of the most appropriate fuel line. Basic tests were conducted within the laboratory. The

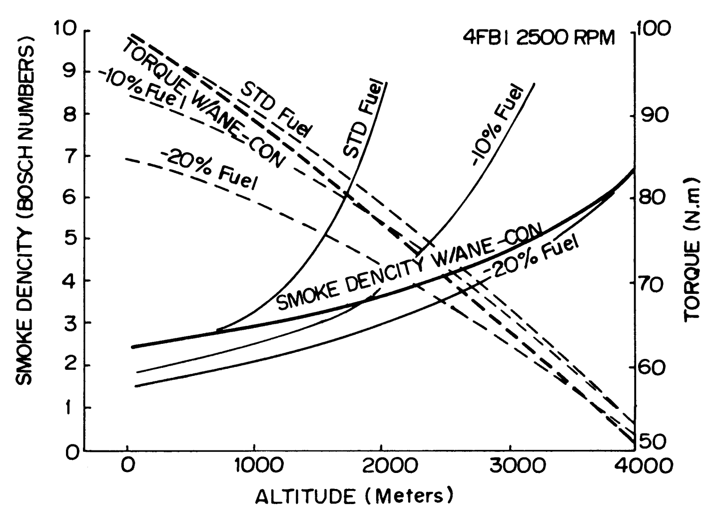


Fig. 57 - Effect of aneroid compensator on engine output and smoke

atmospheric conditions at high-altitude were reproduced for this testing. See Fig. 58. The device shown here functioned to decrease the pressure within the fuel tank by using a vacuum pump. The engine was tested when equipped with this pressure-decreased fuel tank. Based on the results of the testing mentioned above, we adopted a fuel pipe of larger diameter to keep the fuel flow resistance to the minimum. The fuel filter was also re-designed as shown in Fig. 59. The fuel filter of this type is constructed so that the trapping of the air bubbles within the fuel filter can be kept to the minimum.
For evaluating the effectiveness of the improved fuel line against the old type, we made the high-altitude simulation tests within the laboratory and with actual on-vehicle acceleration tests at Mt. Fuji, about 2200 meters above sea-level, both with the engine with this improved fuel line mounted. As you can see in Figs. 60 and 61, the results were very successful. Injection timing was normal, and no surge in vehicle motion was experienced at high altitude.

EASY OPERATION

The ISUZU QSSI SYSTEM - For making this fuel-economy and less-noisy diesel-powered passenger car more popular and acceptable among the customers, it was necessary to have its operation more simplified and easier so that it can be operated like gasoline-powered vehicles. For shortening the engine pre-heat time, the Isuzu Quick Start and Silent Idling System (the ISUZU QSSI System) was newly developed. By adoption of this system, the required pre-heat time was shortened to the level of only 3.5 seconds, as shown in Fig. 62, which is favorably compared to an average of 10 to 30 seconds required by the conventional glow plug system. The glow plug heating by this system lasts for 3 minutes after its preheating has started, improving the engine startability and reducing both engine noise and white smoke generated immediately after the engine start. We already referred to the improvement in the engine startability and in the control of white smoke. We now go into the noise reduction by adoption of this system. The engine noise at idle was reduced by 2 - 3 dB(A) when the temperature of the engine coolant was around 30°C. This is shown in Fig. 63. Function of this after glow mechanism is cut off in either case (1) when the engine coolant temperature comes up beyond 50°C, (2) when 3 minutes has elapsed after the engine gets started, and (3) while the vehicle is in motion. Mechanism of the ISUZU QSSI System is shown in Figs. 64 and 65. It is briefly introduced below. Immediately after the glow plug pre-heating system starts to function, a high electric current flows to the glow plugs, making it get red-hot in a very short time. When the temperature of the glow plugs comes up to the level pre-determined, it is so sensed by means of the resistance value proportionally to change with the temperatures of the specially-selected glow plug heater wire material, and, as a result, the electric circuit is switched to the stabilized current circuit. When the temperature comes down, on the contrary, the electric circuit is switched to the high current circuit again to keep the glow plug temperature on the required level.

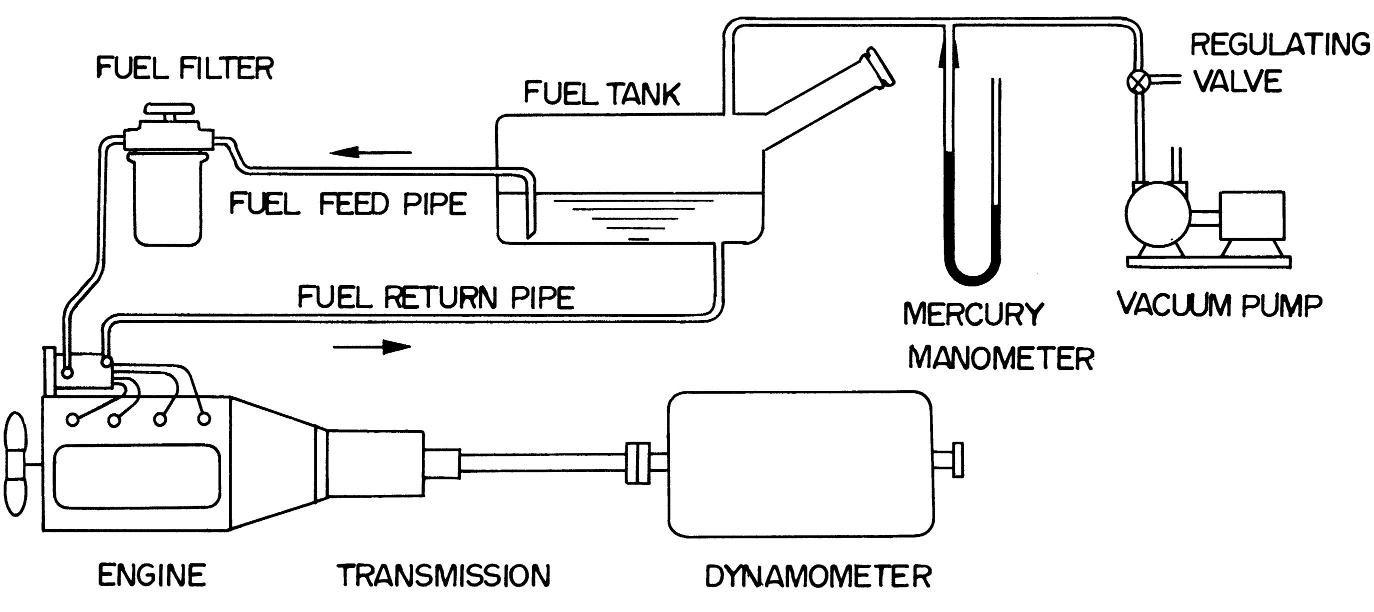


Fig. 58 - High altitude simulation test system

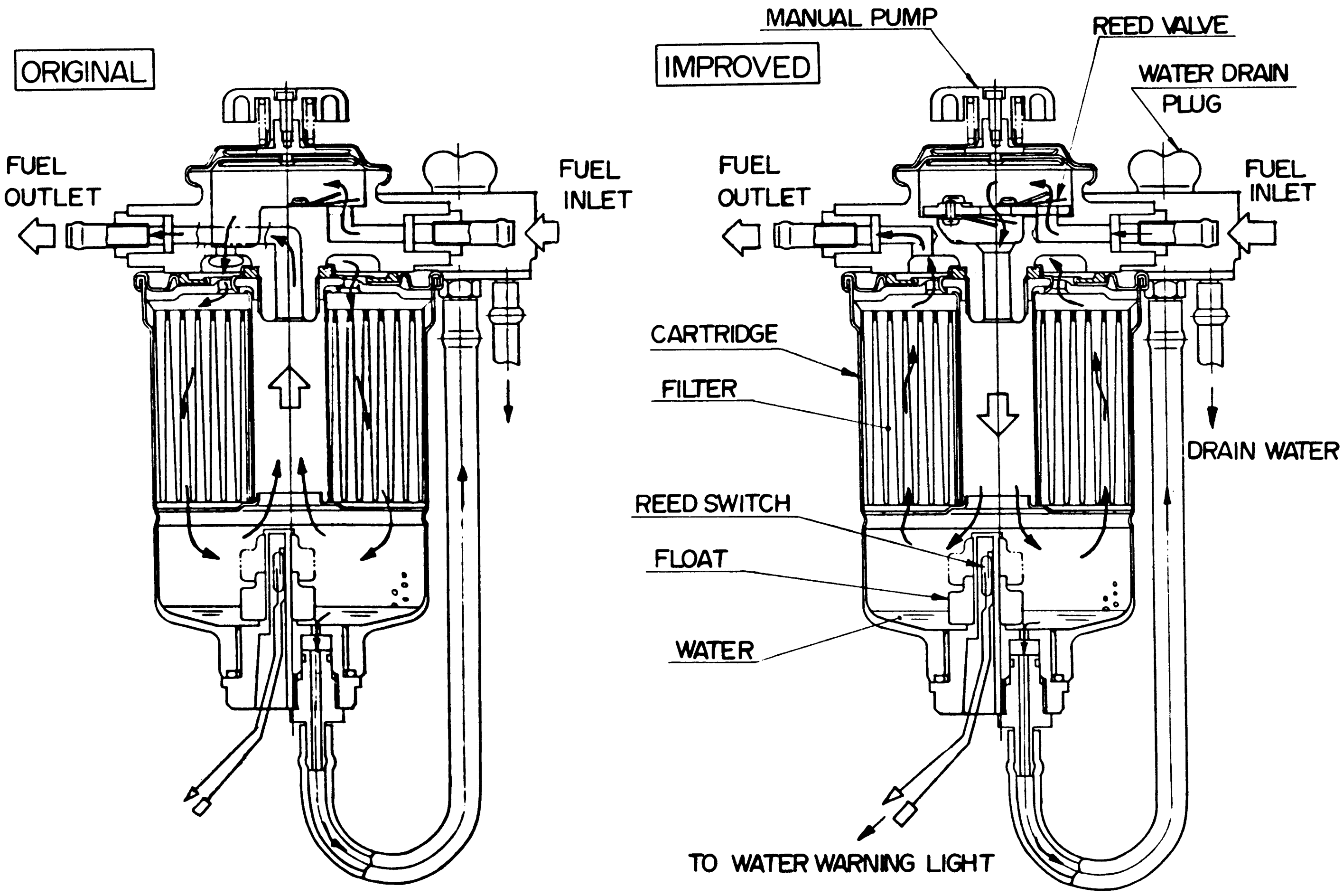


Fig. 59 - Improvement of fuel filter

THE FAST-IDLE SYSTEM - The fast-idle mechanism of the coolant temperature sensing type was adopted. For functioning mechanism, a vacuum actuator was mounted on the injection pump lever. By adoption of this mechanism, below 15°C, the engine speed at idle is increased when the engine coolant temperature is low, securing a smooth running engine at idle.

A WATER SEDIMENTER IN THE FUEL LINE - If the water comes into the fuel tank together with the fuel, it will possibly cause corrosion to develop within the fuel injection system, and also cause the water contained in the fuel to freeze during cold weather. The result will be the clogged fuel line. Several test were conducted. Based on the test results, a most appropriate type of the water separation system was adopted as shown in Fig. 66. This system consists of the tank strainer, the water sedimenter of the within-filter, built-in type with micro-reed switch provided for functioning to actuate the warning system, and the fuel tank drain.

THE FUNCTIONS OF THE WATER SEPARATION SYSTEM ARE AS FOLLOWS:

- The water existing in the fuel tank is almost completely removed by the work of this system. If the water is drained every time when the fuel filter water warning lamp goes on, there will be no water to remain in the injection pump. If the fuel filter water warning lamp comes on and off intermittently, the water must be drained by using the tank drain cock.
- There will possibly be a case, however, where a small amount of water remains within the fuel tank. In that case, the water remaining within the fuel tank will freeze up at cold weather, and the ice particles will start floating around within the fuel tank. In the worst case, such ice particles will be forced into the fuel line, clogging it. In our system, a strainer is provided within the fuel tank. This strainer works to prevent such ice particles from entering the fuel line.

The specifications of the water sedimenter and the fuel strainer are as follows;

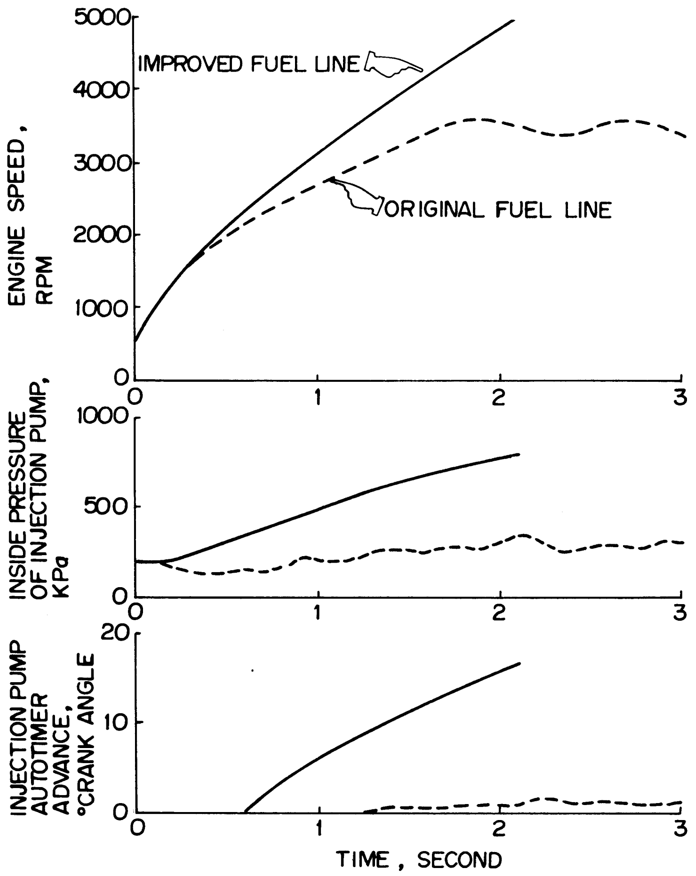


Fig. 60 - Difference of engine acceleration between original fuel line and improved fuel line

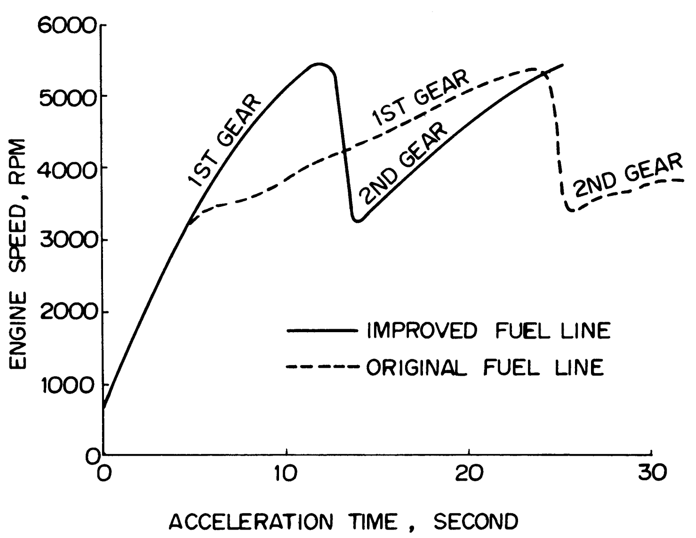


Fig. 61 - Vehicle acceleration test result

Water Sedimenter: Water separation ability at water content 3%: 10~14 PPM (by volume)
Water containing capacity: 150cc.
Indicator lamp on the dashboard comes on when water volume comes up to 80cc.

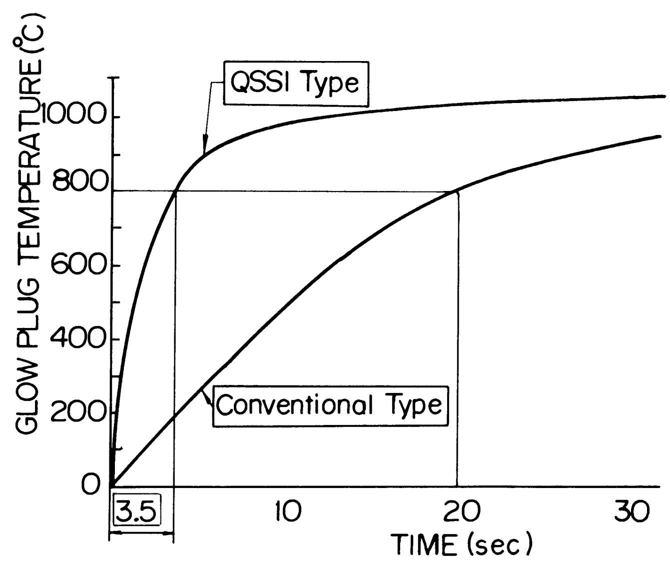


Fig. 62 - Temperature rise characteristics of QSSI type and conventional type glow plugs

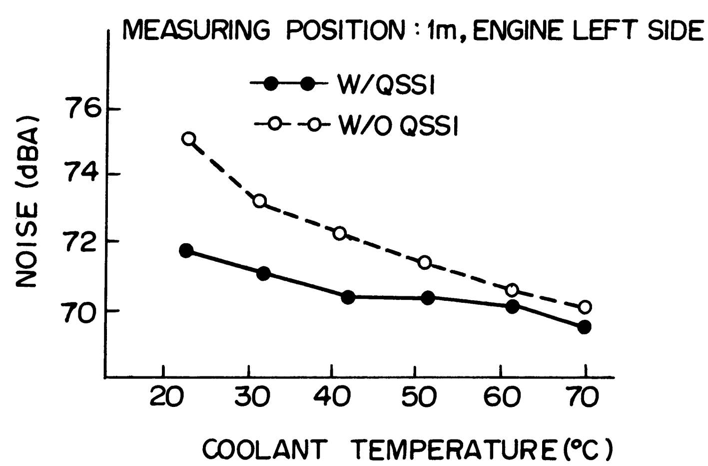


Fig. 63 - Effect of ISUZU QSSI system on engine idling noise

Fuel Strainer : 0.3mm Mesh
Installed in fuel tank.

FOR FURTHER REFINEMENT AS PASSENGER CAR ENGINE

For passenger car use, the 4FB1 1.8-liter diesel engine has come to a level very close to that of its gasoline counterpart. This has been accomplished through the aggressive application of the technology introduced. However, for its further refinement as a fully-compatible outperforming diesel engine for passenger car use, the adoption of several innovative technologies such as turbocharging, electronics, direct injection and use of ceramics is necessary. In the remaining part of this paper, we briefly touch on electronics, direct injection and ceramics.

ELECTRONICS - The ISUZU QSSI System mentioned earlier works by electronic control. Electronics can also be used extensively in

DETECTION OF 900°C

Supply High Current to Glow Plug through Quick Heat Circuit

↓

Increase Resistance of Glow Plug as Temperature rises

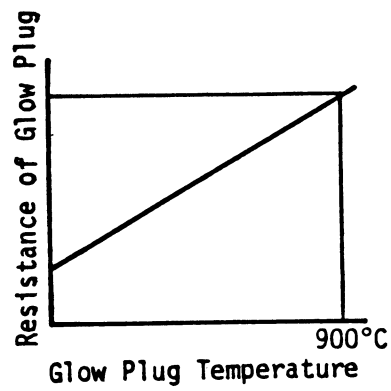
↓

Sense Change of Resistance of Glow Plug by Controller

↓

Switch to Stabilized Circuit for Low Current at 900°C of Glow Plug by Controller

Re-switch to Quick Heat Circuit at 650°C and Lower



Correllation Between Temperature and Resistance of Glow Plug

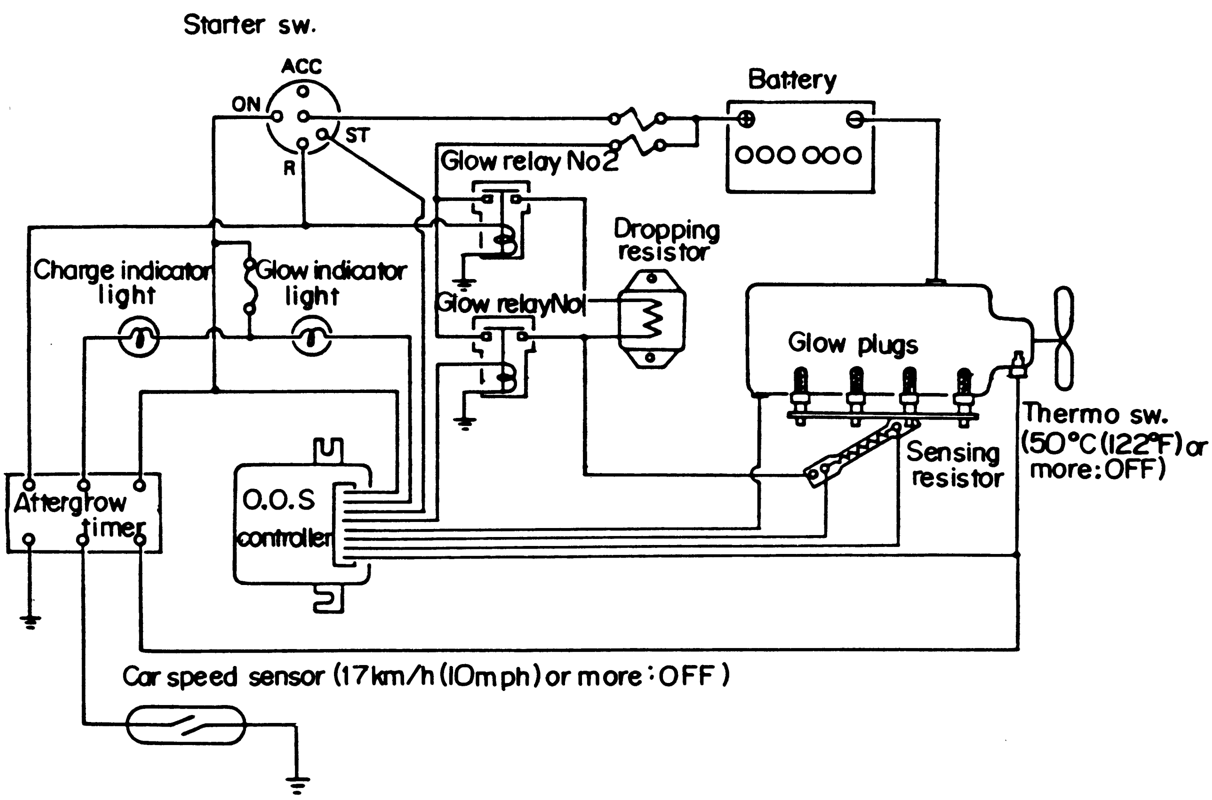


Fig. 64 - Isuzu QSSI system

other areas such as the control of turbocharging, EGR System and engine speed at idle. In place of the mounting of individual electronics control systems for individual functions, a single integrated electronics control system will be preferable. Electronics can also be used for the optimum control of the work of the fuel injection pump. By shifting from the conventional mechanical controls to the electronics controls, we can expect more sophisticated and more sensitive control of the fuel injection which is best able to meet the operational requirements. In the application of electronics, however, the cost-effectiveness would be a matter to consider in advance.

DIRECT INJECTION - The Isuzu Motors first developed a square-type combustion chamber back in 1972. By adoption of this unique design, the engine can offer a high level of fuel combustion performance, despite its relatively small engine displacement, 899cc per cylinder, covering all engine speed ranges from low up to high. Subsequently in May 1981, a 3.3-liter Direct Injection diesel engine was introduced on the market. This engine has per-cylinder displacement of 817cc. Its rated engine speed is 3,600 rpm. Supported by such technological experience so far acquired, the Isuzu Motors will further go into the development of a direct injection diesel engine, further downsizing and further improvement in the high speed range.

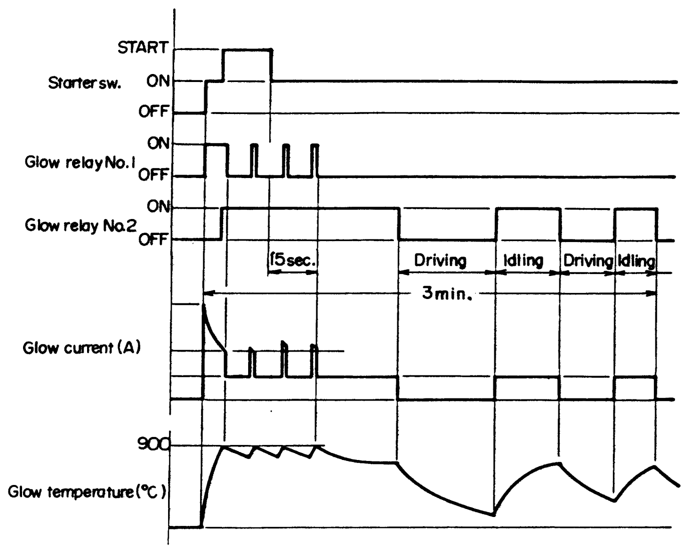


Fig. 65 - Isuzu QSSI system timing chart

USE OF CERAMICS - Ceramics have excellent heat resistivity, adiabatic efficiency, wearability, metal affinity, molding characteristics, etc. Their maximum use in engine components has been being studied. As the first step, a new type of glow plug has been developed. In this glow plug shown in Fig. 67, tungsten lead is directly incorporated in silicon nitride. Ceramics' excellent properties of metal affinity, adiabatic efficiency and heat resistivity made this new design possible. This new type of glow plug has been used on the 82 Model ISUZU 4FB1 1.8-liter diesel engine for a limited specific market. As shown in Fig. 68, by adoption of this new type of glow plug, less pre-heating time and easier engine start comparable to gasoline engines have been obtained.

The research on more extensive use of ceramics is going on, too. Included are their use in the hot plug of the swirl chamber, in the particulate trap, etc. The adoption of the ceramics in these parts, however, may possibly take much time before their quality stability, performance, thermal and mechanical strength, reliability, and production cost requirements are all cleared to a satisfactory extent.

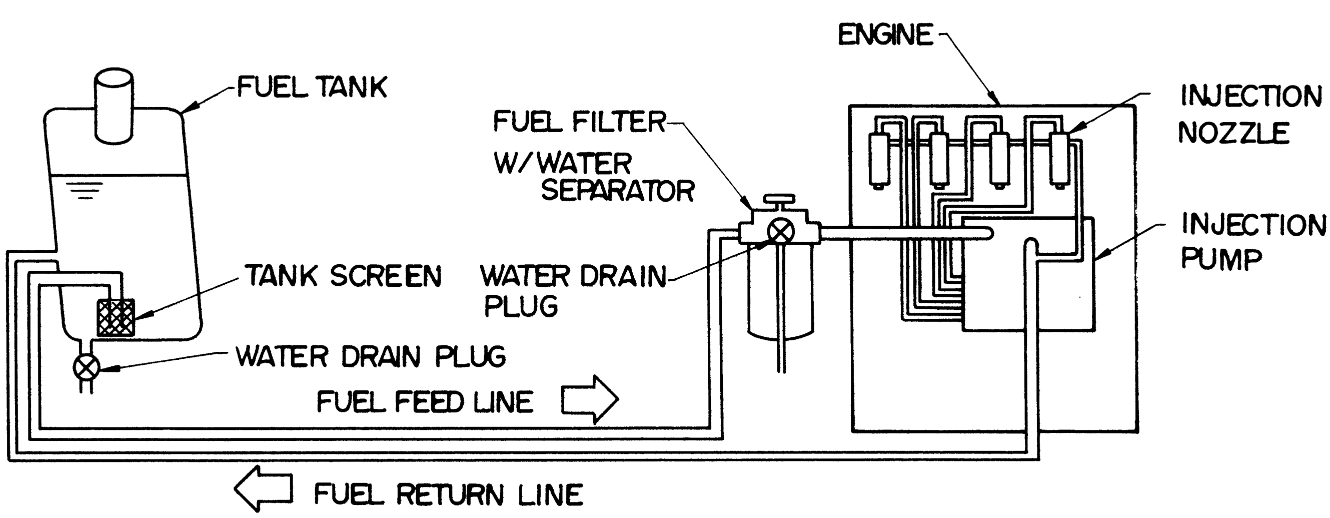


Fig. 66 - Fuel system of Isuzu I-Mark diesel

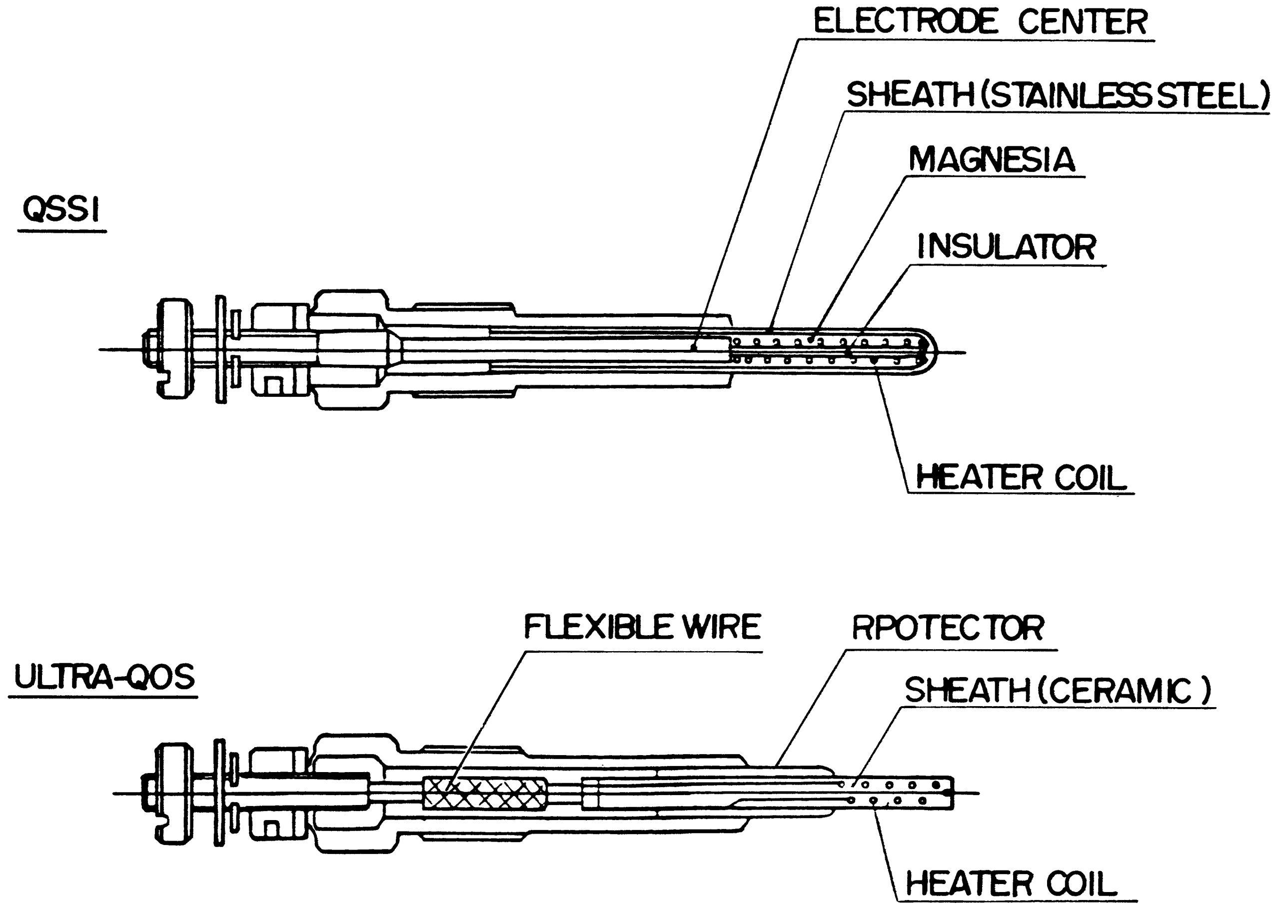


Fig. 67 - Structure of glow plug, QSSI and Ultra-QOS

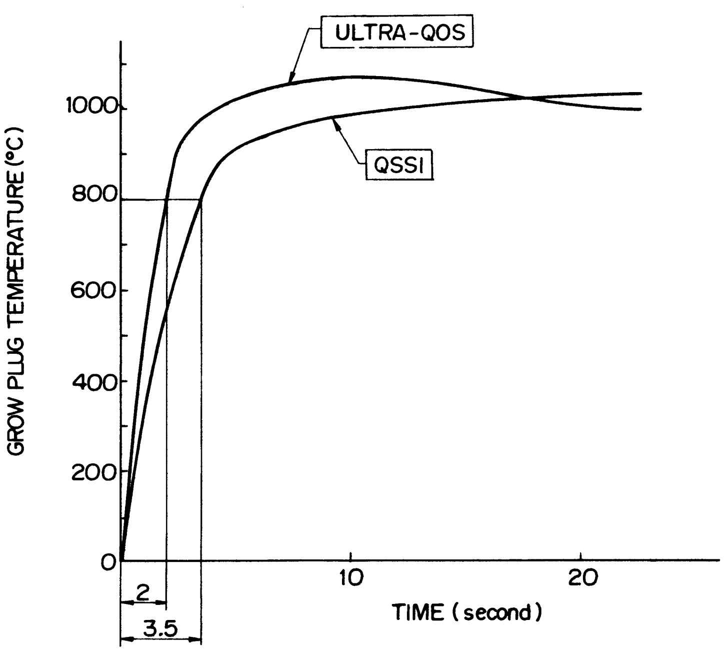


Fig. 68 - Temperature rise characteristics of Isuzu Ultra-QOS type and QSSI type glow plugs

SUMMARY

As introduced, the Isuzu 4FB1 1.8-liter diesel engine is an engine featuring a high level of fuel economy and a satisfactory level of product compatibility to meet the U.S. requirements, both environmental and operational in-field use. We believe that this engine is qualified as an engine for passenger car use, answering the expectations of the U.S. customers fully. The Isuzu Motors Limited will continue its efforts for the advancement of innovative automotive technology, reflecting its outcome on this engine for its further refinement in fuel economy, ease of operation and other aspects of performance with no cost penalty thereby.